INFLUENCE OF ENVIRONMENT ON A FAN DYNAMOMETER

BY

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1916

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Characteristics of and
influence of environment on

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CHARACTERISTICS OF AND INFLUENCE OF ENVIRONMENT ON A FAN DYNAMOMETER

A THESIS

PRESENTED BY

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TO THE

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FOR THE DEGREE OF

BACHELOR OF SCIENCE IN MECHANICAL ENGINEERING

HAVING COMPLETED THE PRESCRIBED COURSE OF STUDY IN

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+:+:+:+: PART ONE :+:+:+:+

I N T R O D U C T I O N.

oOo

Object of thesis.

-

Characteristics of various dynamometers.

-

Equations for determining the horsepower of a fan dynamometer.

I N T R O D U C T I O N.

-oOo-

The object of this thesis is to determine the characteristics of and the influence of environment on a fan dynamometer. The horsepower of an engine, when measured by a fan dynamometer, is dependent on an equation which, it is supposed, varies with the pressure of the air against the fan blades, the barometric pressure, the temperature of the air, and the relative humidity. The environment of the dynamometer also influences results, whether the blades are entirely open to the air, or partially or wholly enclosed, and whether objects are in proximity to the dynamometer. The object of this test is to design and build a fan dynamometer and to investigate these characteristics of this dynamometer, developing, if possible, an equation involving their use.

A dynamometer which would be of the greatest commercial value must be cheap in first cost, accurate, must have a wide range in speed with a constant torque at each speed, must provide a means for the torque's dropping off in case the speed decreases, must be easily manipulated, and should require no calibration. No dynamometer as yet developed will realize all of these conditions.

A prony brake has the characteristic that the torque is independent of the speed. Thus, if the speed of the engine momentarily reduces, caused perhaps by the missing of an explosion, the brake is liable to grip the shaft and stop the engine. This brake requires a water ccoled flywheel on which it is placed, and even then it is hard to prevent excessive heating.

This heating results in a variable torque resistance, which makes it difficult to maintain a constant load on the engine.

It is almost impossible to hold a constant torque for any length of time with this dynamometer at high speeds. The prony brake is simple in construction and cheap in first cost, and is accurate.

There are two or three electrical absorbtion dynamometers. They are all high in first cost and rather complicated. All of them have to be calibrated for mechanical efficiency at various loads and speeds, and even then many errors enter into the work. The labor in conducting the test and in making the computations is large. In this type of dynamometer the torque varies approximately as the square of the speed.

The electrical cradle dynamometer obviates these difficulties, but represents a high initial cost.

The characteristic of the water brake is that the torque rises with the speed. On account of its bulk, it is unsuitable for low speeds. Its horsepower

is represented by an equation, the functions of which vary with the speed, and temperature, density, and other physical properties of the water used.

The water brake is exceedingly cheap and easy to operate.

When combined with a free floating casing, the direct reaction can be obtained on a pair of scales as with the prony brake. This makes a precise method of measuring power, and the torque or horsepower characteristics of the dynamometer cause a sweet operation of the automobile type of engines.

The reaction dynamometer would be the ideal dynamometer for all classes of testing work if it were not for the fact that the engine has to be loaded in some manner before it will develop horsepower.

The fan dynamometer is attractive because of its adaptability, cheapness, and simplicity. Its great disadvantage lies in the fact that it is very susceptible to changes of air pressure, temperature, humidity and environment. The torque in this dynamometer varies as the square and the horsepower as the cube of the speed, thus making it especially adapted to

high speed work.

A fan dynamometer is a fan whose blades are placed at right angles to the plane of rotation so that they will offer resistance to the air as they revolve. A cross arm is placed on the extended crankshaft of an engine at right angles to the plane of rotation of the shaft. A blade, or plate, is placed on the outer extremity of each arm, so that its surface offers resistance to the air as the cross arm is rotated by the engine. The arrangement of all fan dynamometers is on this order.

There are several equations in use for the determination of the horsepower absorbed by a fan dynamometer. In all of them there is a constant determined by experiment.

The following is from Horseless Age, May 20th, 1914, page 808. When a body is moved through the air in a straight line, the power required to overcome the resistance of the air is proportional to

the third power of the velocity of the body. The power absorbed at a given speed, with the plates at different distances from the axis, varied as if the resistance on elements of the plate was proportional to the 2.5 power of the radius of rotation. In figure 2, page 8, let dr be such an element whose radius of rotation is r. The power absorbed may properly be assumed to be proportional to the width of the plate. Let n represent the R.P.S. of the fan, and c a constant denoting the power absorbed by the air resistance on a unit area located at a unit distance from the axis of rotation, at unit speed of revoltuion. Then the power abosrbed by the element dr is

$$H.P. = n^3 c w r^{2.5} dr$$

and the power absorbed by the whole plate is

$$H.P. = \int_{r_2}^{r_1} n^3 c w r^{2.5} dr$$

$$= \frac{n^3 c w}{3.5} \left(r_1^{3.5} - r_2^{3.5} \right)$$

Since there are always two plates, the total

power absorbed is

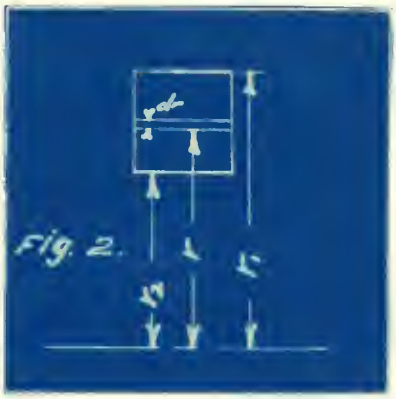
$$H.P. = \frac{2c}{3.5} n^3 w \left(r_1^{3.5} - r_2^{3.5} \right)$$

From data available, it was found that

$$\frac{2c}{3.5} = \frac{1}{68 \times 10^6}$$

$$H.P. = \frac{n^3 w}{68 \times 10^6} \left(r_1^{3.5} - r_2^{3.5} \right)$$

To be applied first to the plate and then to the beam.



n = R.P.S.

w = in inches

r = in inches

The next equation is from the Transactions of the Society of Automobile Engineers, Book 2, vol.8 pp 110-134. In all cases of fan work it is known that

$$H.P. = CN^3$$

where $N = \frac{R.P.M.}{1000}$

$$C = 8\pi^3 a A R^3$$

Where a = resistance constant of air per unit area

A = area of plates in sq.cm.

R = radius of outer edge of plate in dec.

From experimental date for square plates it was found that

$$8\pi^3 a = \frac{1}{4010}$$

H.P. for square plates

$$= \frac{AR^5 N^3}{4010}$$

Changing dimensions to inches so that

A = area of plates in sq.in.

R = radius of outer edge of plate in inches.

$$N = \frac{R.P.M.}{1000}$$

$$H.P. = \frac{AR^5 N^3}{37830}$$

In the same article is given a formula without derivation.

$$H.P. = C\ m\ N\ n^3 \left[(B-b)\ (R^4-r^4) + bL^4)\right]$$

Where C = a constant

m = specific gravity of air

N = number of blades

n = R.P.M.

B = width of blade in inches measured parallel to shaft

b = width of beam in inches measured parallel to shaft

R = radial distance of outer edge of blade from center of shaft in inches.

r = radial distance of inner edge of blade from center of shaft in inches.

L = radial distance of outer edge of beam from center of shaft in inches.

From experiment it was found that

$$C\ m = 6.9 \times 10^{-15}$$

The dimensions are shown in figure 3, page 11.

The following is from The Automobile, April 6, 1916, page 627. No derivation is given. It is good only for square or round blades.

$$H.P. = \frac{u\ N}{63000}$$

$$u = u_a + u_b$$

$$u_a = .35 \times 10^{-8}\ W N^2 r_a^2 a^4 t\ W$$

$$u_b = K N^2 r^3 s^2 W$$

Where

W = density of fluid in which fan rotates in lb./cu. ft.

= 62.33 for water at 50° F.

= 1.347 x barometric pressure in in. of Hg.

u_a = torque due to arms alone in in.lb.

u_b = torque due to blades alone in in.lb.

N = R.P.M.

s = length of side of square blade or diameter of circular blade in inches.

r = external radius of fan in inches

r_a = radius of arms in inches measured to the inside of the blades.

t = thickness of arms in inches.

K = a coefficient depending upon the value of $\frac{r}{s}$ and the shape of the blade for fans rotating in free space.

σ = a coefficient depending on the section of the arms.

All of the above formulas have one or more constants dependent on experiment. A rational formula would be far more dependable and accurate. In connection with the results of the test, there is an attempt to work out such a formula.

+:+:+:+:+PART TWO+:+:+:+:+

DESCRIPTION OF APPARATUS.

oOo

Introduction.

The reaction dynamometer.

Theory of the reaction dynamometer.

The Engine.

The fan dynamometer.

Design of Keys.

Design of Bolts.

Balancing of Engine

Incidentals.

DESCRIPTION OF APPARATUS

-:-:-:-

The set up of the apparatus was accomplished in this manner. The engine was placed in a cradle or reaction dynamometer. The engine itself drove the fan dynamometer. Since the fan brake loaded the engine, the reaction dynamometer gave a precise method of determining the horsepower. With the horsepower already determined, the constant of the fan dynamometer could be very easily determined.

A photograph of the thesis is shown in figure 1, page 61.

The whole apparatus was set on a base furnished by Joseph Tracy, New York. This base supports a reaction dynamometer cradle. Two bearings, one at one end of the base, and another near the other end, support the cradle which is of sufficient size to mount the engine. The cradle is hung in these pedestals with

ball bearings in such a way that it will tip very easily. Suitable stops prevent excessive tipping.

The engine is placed in this dynamometer frame, and firmly fastened to it by means of equipped with lock washers. Near one end of this cradle dynamometer frame a threaded shaft extends downward nearly to the base.

This shaft carries adjusting weights which permit the center of gravity of th-e whole piece of apparatus to be raised or lowered. Under normal operation, the center of gravity is adjusted to the center of the pedestal bearings.

Arms extend, one from each side of the dynamometer, by which it is balanced and the torque is measured. Each of the pedestal bearings has a hole through it on a line with the engine crankshaft. The engine crank extends through the front hole and the fan dynamometer shaft through the rear one.

The engine crank extends through the front hole

and the fan dynamometer shaft through the rear one.

The theory of the reaction or cradle dynamometer is very simple. When the engine is developing horsepower, it exerts a force or action. The reaction is felt on the dynamometer so that it tends to tip in the opposite direction. See figure 5, page 16. There is an arm extending from the side of the reaction dynamometer outward.

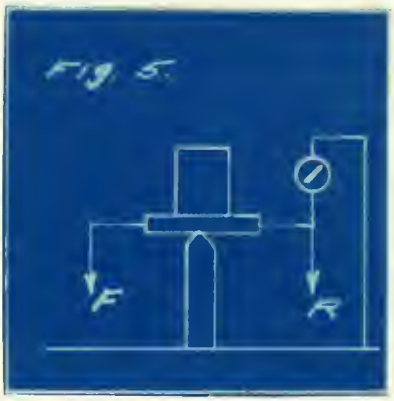


Fig. 5.

As this reaction is felt, the arm will tilt down and register a certain weight on the scale.

Now H.P. = $\frac{FS}{33000}$

Where F = force

S = $2\pi a N$

Where a = length of arm from center of rotation.

N = R.P.M.

This moment = Fa = M.

H.P. = $\frac{2\pi MN}{33000}$

The length of the arm from the point where it is fastened to the scale to the center of the rotating shaft is

$$a = 31.50''$$

$$= 2.62'$$

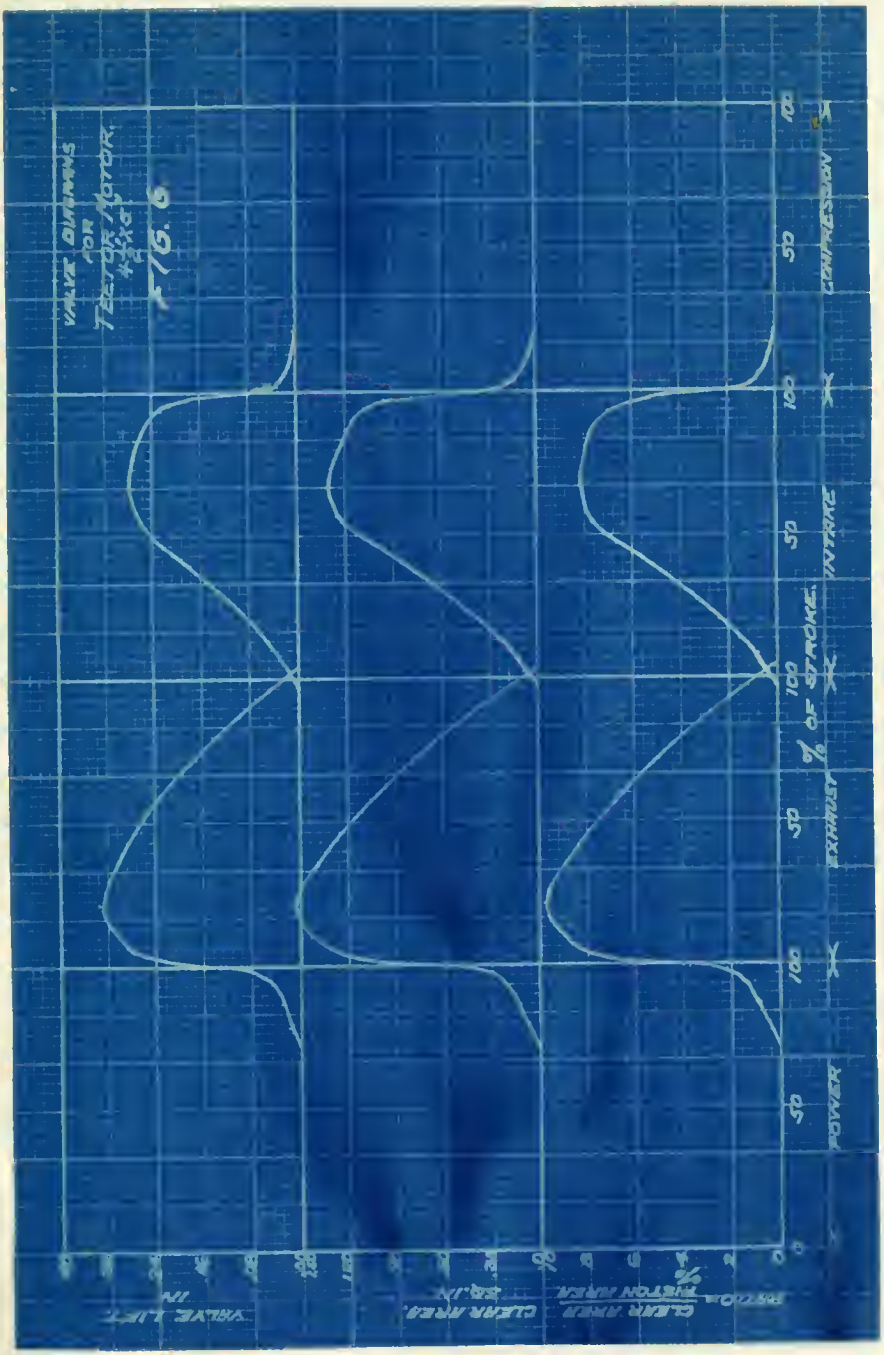
Then H.P. $= \frac{MN}{5250} = \frac{WN}{2000}$

Where W = weight registered on scale.

The engine used in this thesis is a four cylinder four cycle 4-1/2" x 5" T head Teetor automobile motor .

The cylinders are cast en bloc and a three bearing crankshaft is used. Lubrication is by the splash system to the pistons and connecting rods, and by a forced system to the half time gears and main bearings. A Bosch two spark magneto and the contrifugal circulating pump are driven from a jackshaft geared through the half time gear operating the exhaust camshaft.

A 1-1/4" model A Motsinger carburetor is used.


VALVE DIAGRAMS
FOR
TEST OF MOTOR
FIG. 6
POWER
EXHAUST
% OF STROKE
INTAKE
COMPRESSION
0
50
100
CLEAR AREA
PISTON AREA
CLEAR AREA
VALVE LIFT
IN

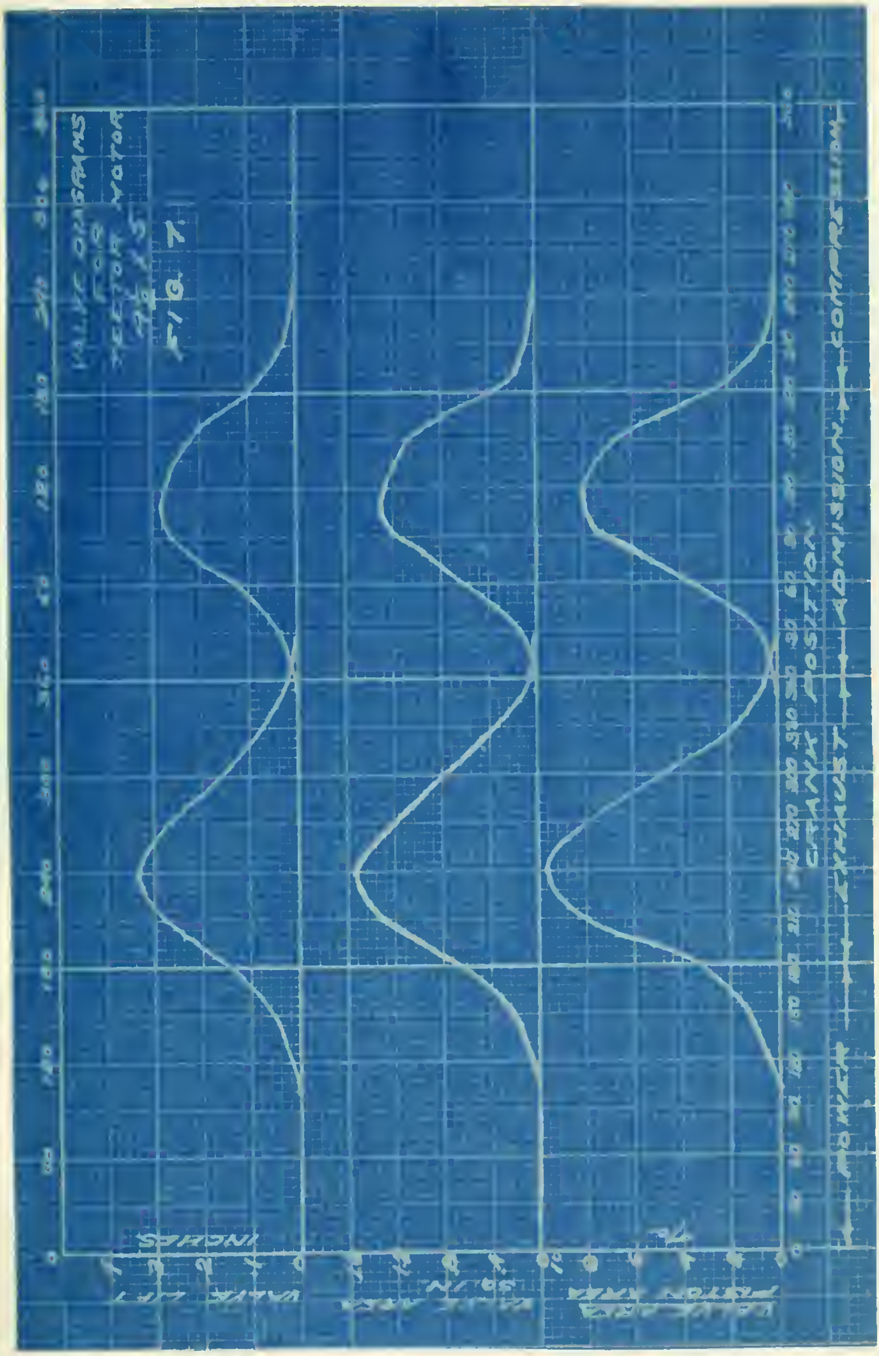


FIG. 7
CRANK POSITION
EXHAUST — ADMISSION — COMPRESSION
VALVE LIFT
INCHES

The fan is belt driven from a pulley on the over hanging end of the jackshaft. An Overland-Kinsey radiator is used to cool the water. Diagrams of the valve action referred to the angular crank positions and to the percents of stroke, are shown in figure nos.6 and 7 on pages 18 and 19. An interesting feature came up in connection with the adjustment of this engine.

It was discovered that poor compression in the cylinder was due to the fact that the fibre-distance piece between the cylinders and crank-case was dried out, and that the crankcase had settled a little. This decreased the valve stem clearance until the valves were held open at all times. When the bolts holding the two together were tightened and the valve stem clearance adjusted, the compression was normal again.

The fan dynamometer itself was entirely designed and built by the authors of this thesis.

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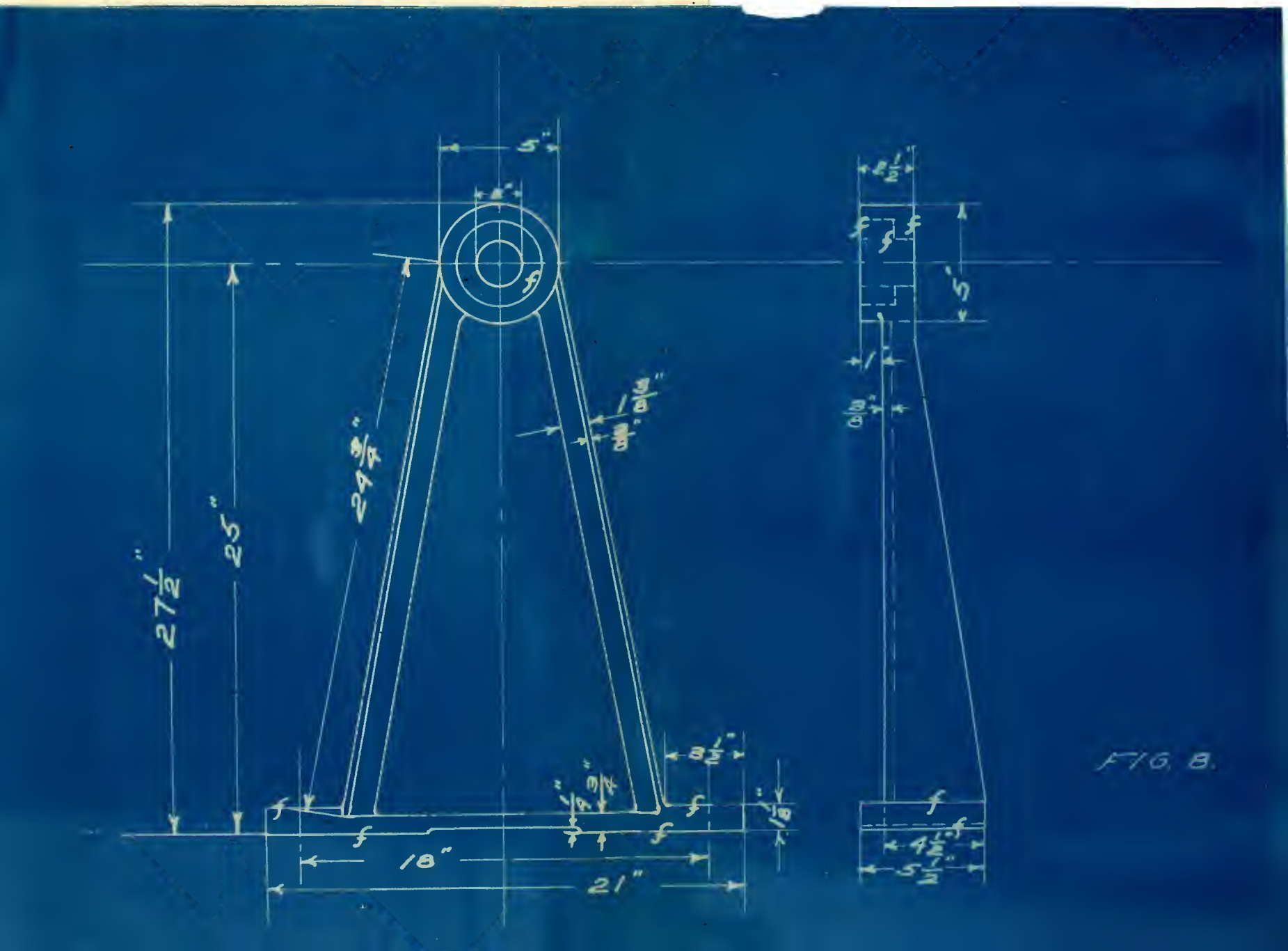


FIG. 8.

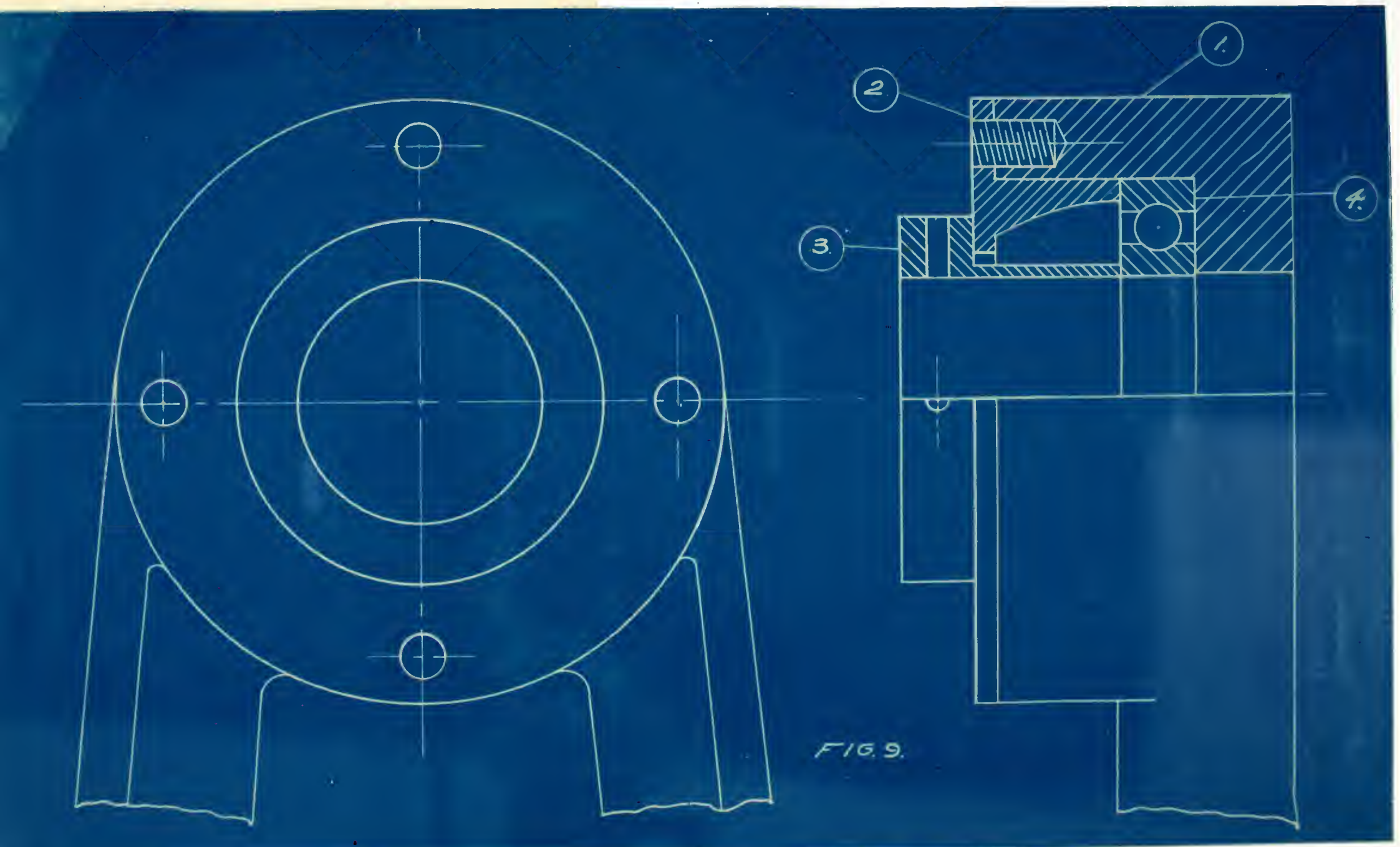


FIG. 9.

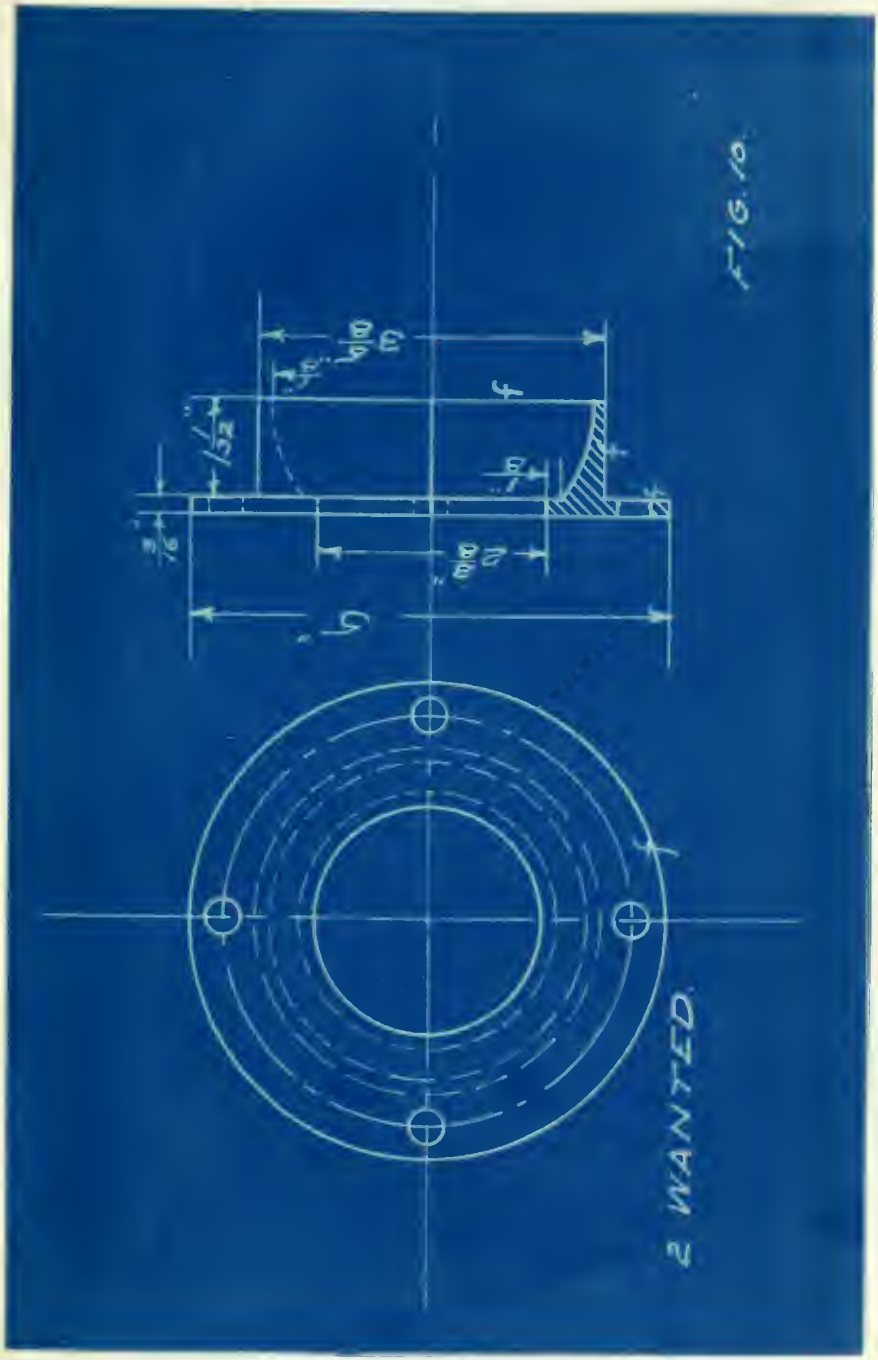


FIG. 10
2 WANTED.

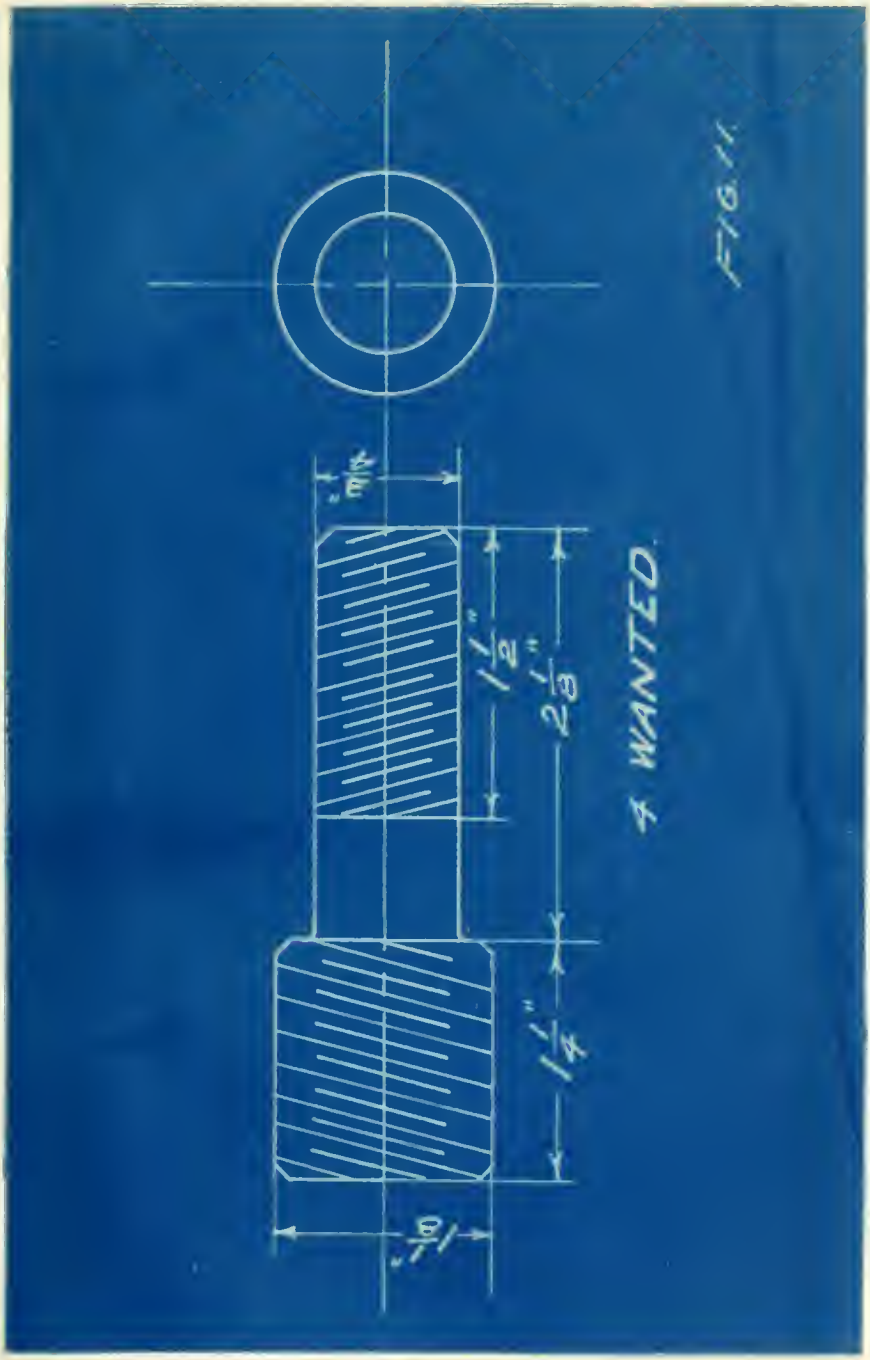


FIG. 11.

In the base of the apparatus had been drilled four holes for a fan dynamometer. They were in the end opposite to the reaction dynamometer. The width of the bottom of the bearings supporting the dynamometer shaft was approximately fixed, as was also the distance between the bearings. The bearings were designed as shown in figure No.8 page 21. They were to be stout enough so that they would not vibrate, and there was to be as much room between them as possible for the blades. A pattern for these bearings was made and two castings were obtained from it. The upper part of each of the bearings was bored and counterbored as shown in figure 9, page 22, to accommodate a Hess-Bright ball bearing. The fit was to be snug so that there would be no chattering of the ball bearing in the castings at high speeds. A special cap was made, shown in Figure 10, page 23, and fitted to the bearings, see figure 9, so that it bolted on one face of the casting.

It pressed against the outside housing of the

ball bearing, holding it tightly in place, while at the same time it allowed the inner housing to revolve with perfect freedom. Small slots were cut in the base of the castings so that easy adjustment could be made when it came to setting up the apparatus. The holes in the base of the apparatus were tapped out and special bolts, see figure 11, page 24, were made to hold the casting to the base. When it came time to set up, shims were made of such a height that the fan dynamometer shaft lined up with the engine crankshaft. The flywheel of the engine had cut in it a space for a cone clutch.

An extra part to the engine was a cover to this flywheel, see figure 12, page 28. A row of holes on this cover gave a means for bolting it to the flywheel. Near the center of the face of this cover was another set of holes. It was decided to use pins in these holes and to drive the dynamometer shaft through a leather coupling.

The dynamometer shaft was therefore made of such a length that there was just clearance between it and this flywheel cover on the one end and that it extended about two inches beyond the bearing on the other end, see figure 13, page 29. The ball bearings had an internal diameter of 50 mm, or 1.9685". The shaft was therefore cut to that diameter and a very close fit was made, by grinding, between the shaft, and the two ball bearings, and the arm, which will be described a little later. The flywheel end of the shaft was cut down so that a flange could be fitted to it, which would connect it to the flywheel through the leather.

A keyway was cut in the shaft for the flange, and another near the center of the length of the shaft for the blade arm. The latter keyway was made long so that the arm could be shifted along the shaft. The flange coupling was made of a nickle steel drop forging.

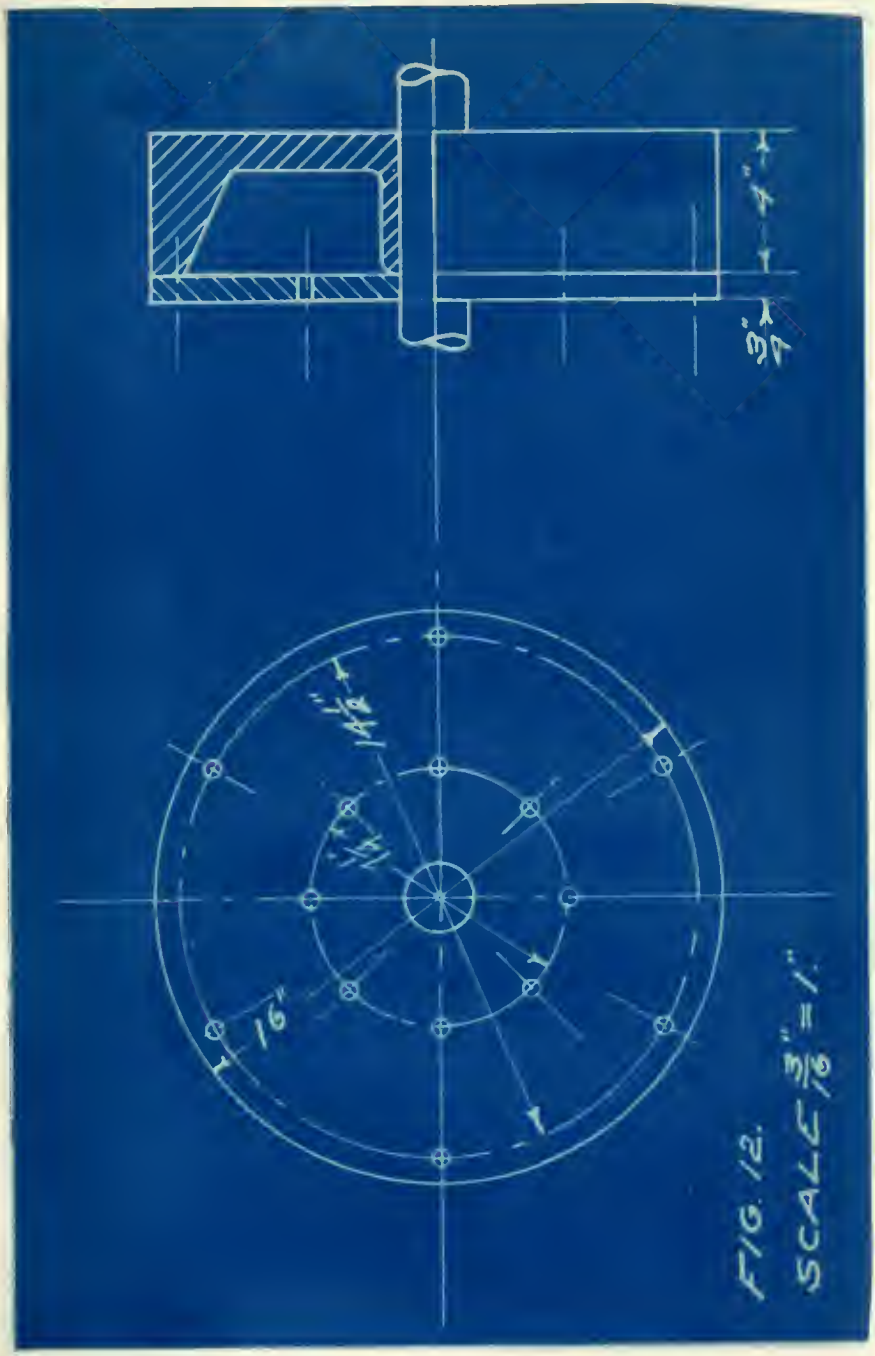


FIG. 12.
SCALE 3/16" = 1"

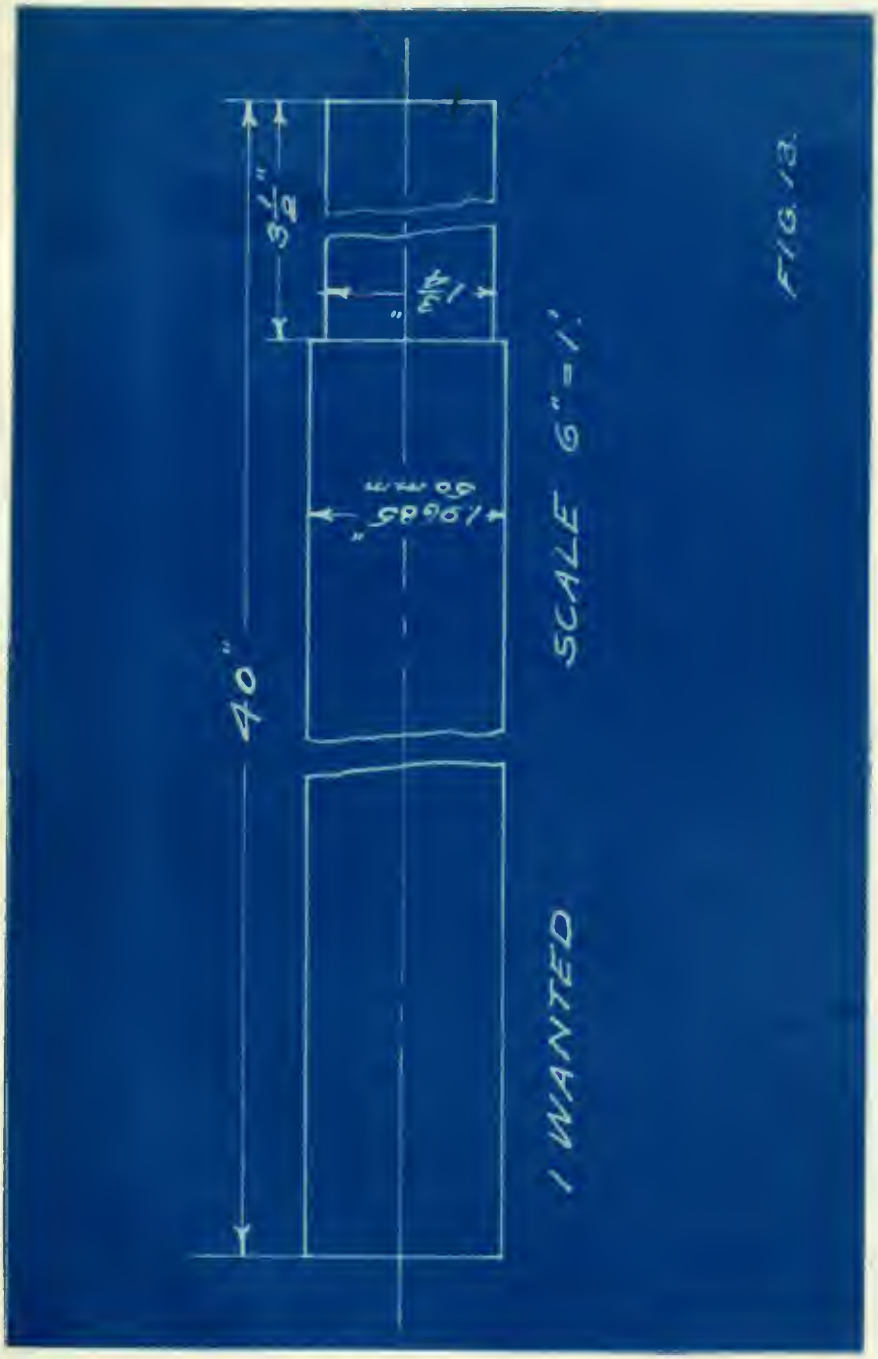


FIG. 13.

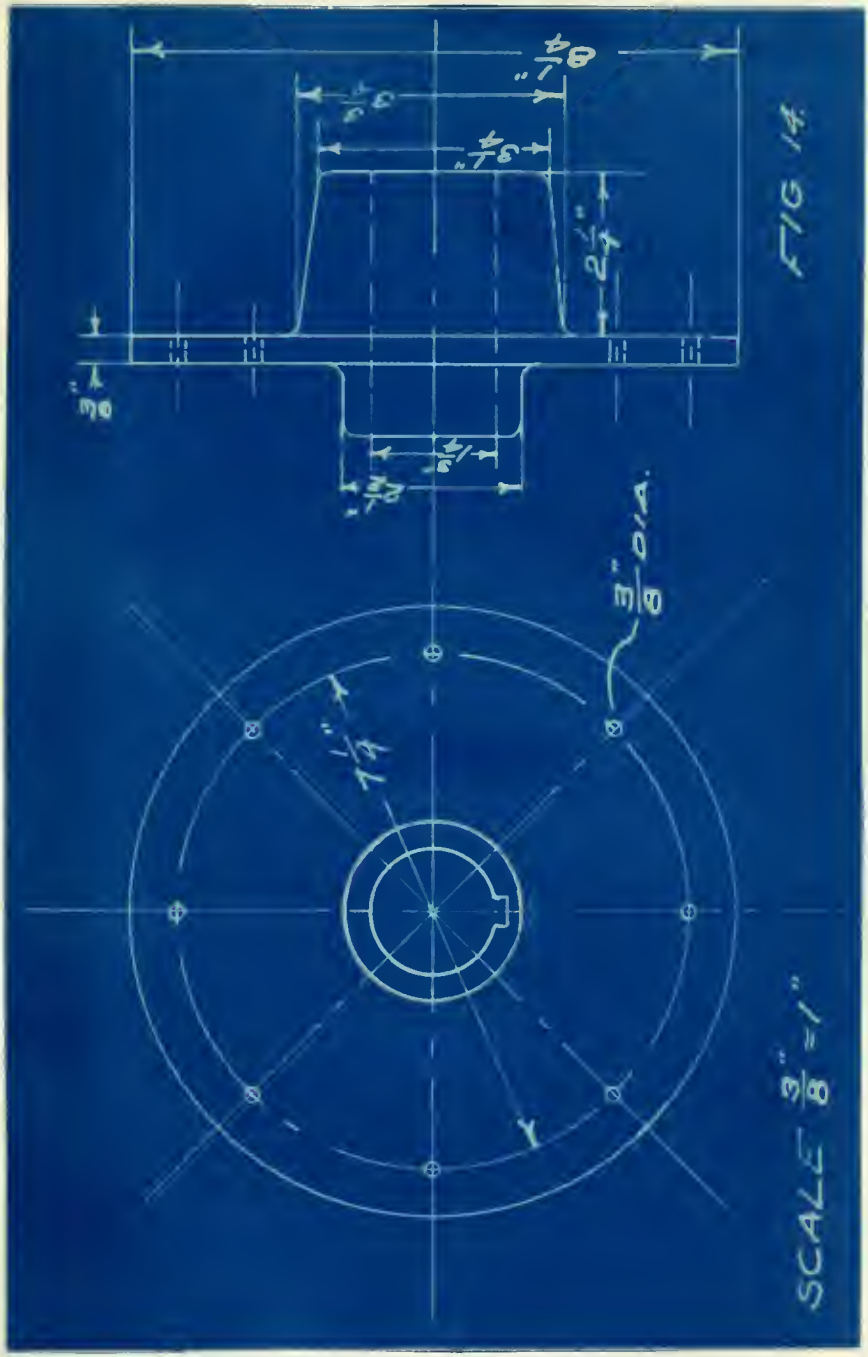


FIG 14
SCALE 3/8" = 1"
3/8" DIA.

Its dimensions are shown in figure 14, page 30. A hole was drilled,bored, and reamed for the shaft. Along the outside of the face of the flange a set of holes, corresponding exactly in size and position to the inner set on the flywheel cover, was bored.

A keyway was cut in the flange to correspond to the keyway in the shaft. Two set screws, set at right angles to each other, were used to hold the flange to the shaft. One set screw fitted over the key. A set of pins, see figure 15 page 33, were made, and one was placed in each hole, both on the flywheel cover and on the flange. Three rawhide discs, see figure 16, page 34, were cut out to fit these pins, a pin from the flange alternating with a pin from the cover around the circumference. The centers of the leather strips were cut out to allow room for the hub of the flange. The flange was then placed very close to the cover so that there was

just clearance between the two sets of pins and the faces opposite them. This gave a neat, flexible, and secure method of transmitting the power from the engine to the fan dynamometer. When the dynamometer shaft was connected to the engine crankshaft, it was found that the engine crankshaft was a little higher than the center of the hole through the reaction dynamometer bearing. As the fan dynamometer shaft was very little smaller in diameter than this hole, it scraped against the top of the hole. The top of the hole was, therefore, chiseled and filed out to make the extra clearance room. Collars, see figure 17, page 75, were made for the shaft to prevent its sliding along its length so that the coupling would become loose, or perhaps, the arm move over so that the blades would strike the bearings. The collars were set screwed to the

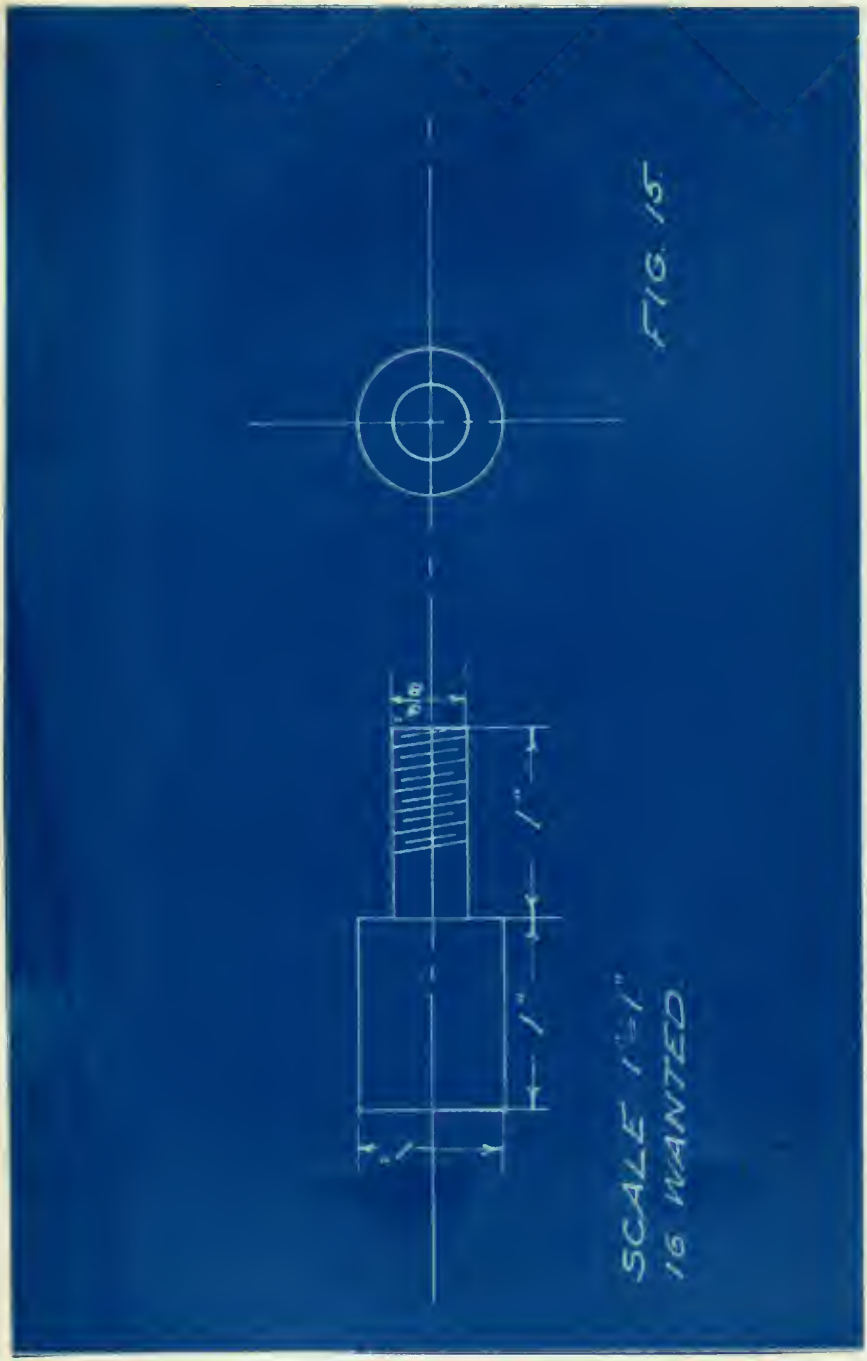


FIG. 15

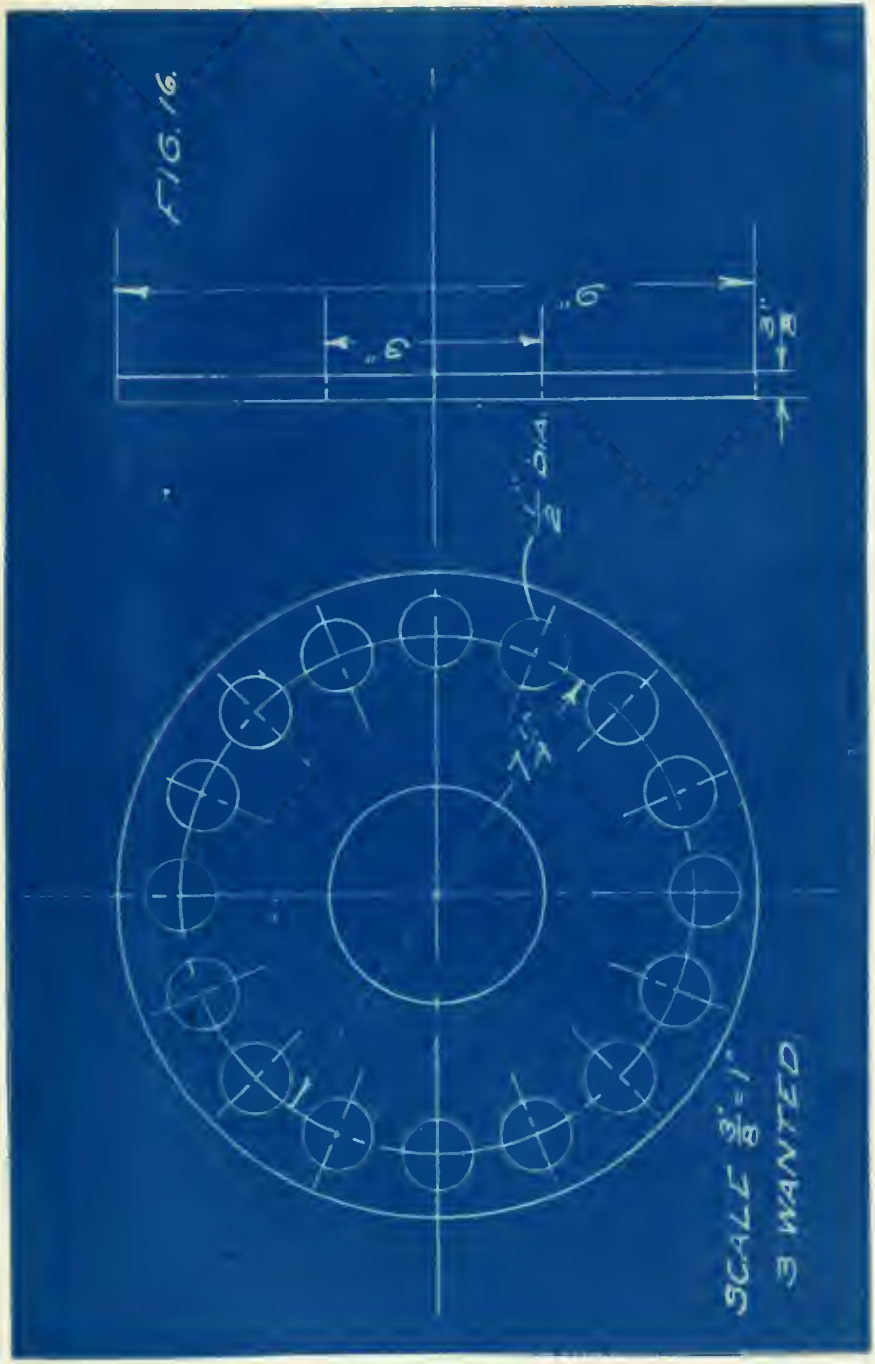
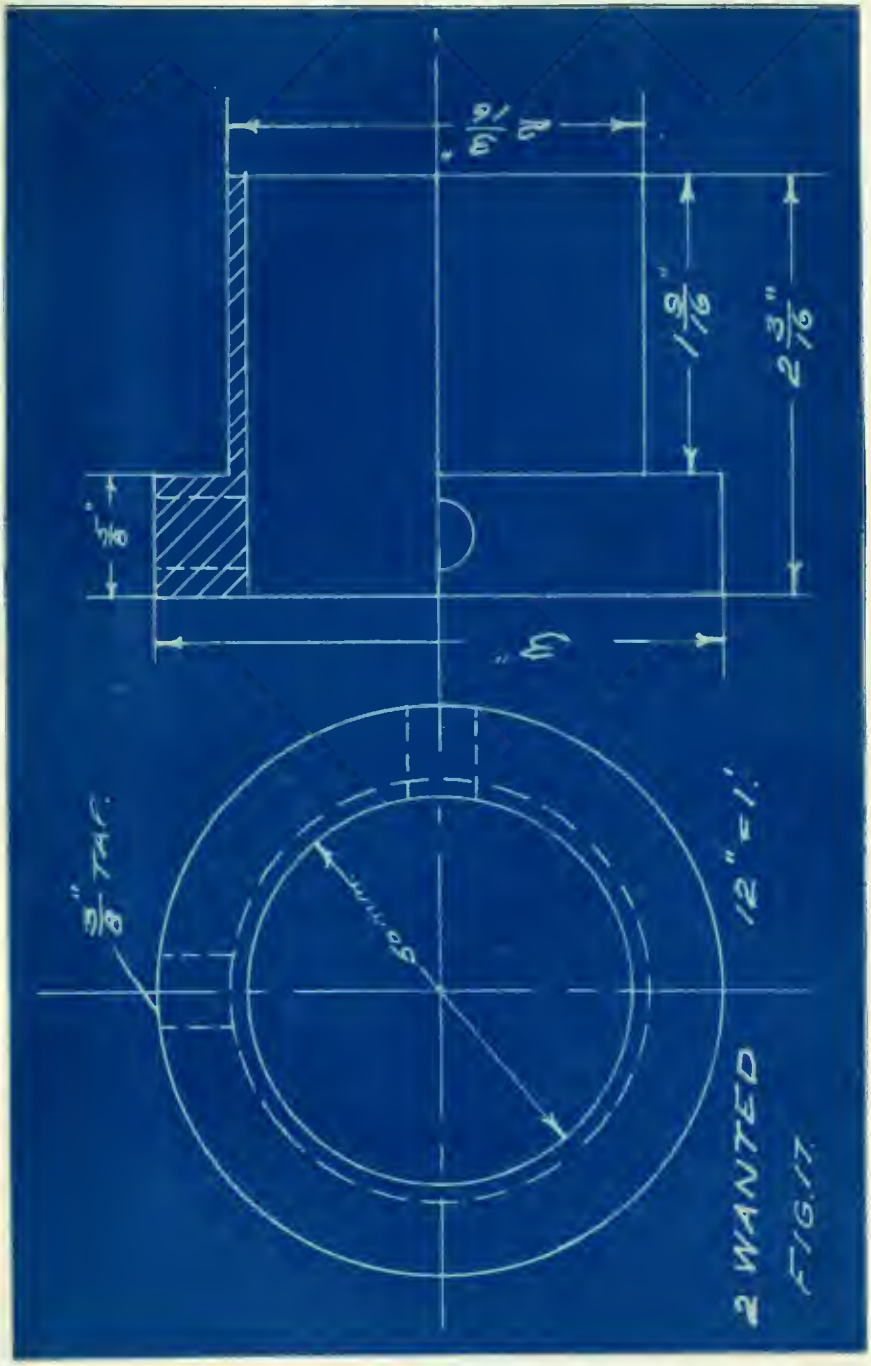


FIG. 16.
3"
6"
2"
1/2" DIA.
1 1/2"
SCALE 3/8" = 1"
3 WANTED


2 WANTED
FIG. 17.
12" = 1'
3/8" TAP.
1 9/16"
2 3/16"
1/4"

shaft with two set screws at right angles to each other. These collars fitted up against the inside housing of the ball bearings, see figure 9, and thus held the shaft stationary lengthwise, since there was one collar for each bearing. The arm was forged out of nickle steel. It was then machined all over, see figure 18, page 38, and balanced. Holes were drilled along the face of each arm so that the blades could be moved in or out along the arm. The arm was drilled and fitted to the shaft as snugly as possible. Two set screws were placed on the arm on the opposite side of the keyway, which was cut in it, so that they could help balance up the key. Keys were made for both the arm and for the flange, see figure 19, page 39. The computations for their design is given later.

The blades, or plates, see figure 22, page 44, were fastened to the arm by means of angle

irons. Two angle irons were used with each plate. The dimensions of the angle irons are shown in Fig.20,page 40, the method of connection to the arm and plate in figure 21, page 43. One face of the angle iron fitted up against the face of the arm. This face of the angle iron had three holes drilled in it to fit exactly any three corresponding holes drilled in the face of the arm. On the other face of the angle iron were drilled two holes for the plate. One angle iron fitted against each face of the arm and one bolt went through the two angle iron pieces and the arm, thus holding the three securely together. The first set of plates, see figure 22, were cut of such size that they would just clear the bearings. Four holes were drilled in each of them for a means of fastening them to the angle irons. This completes the construction of the fan dynamometer itself.

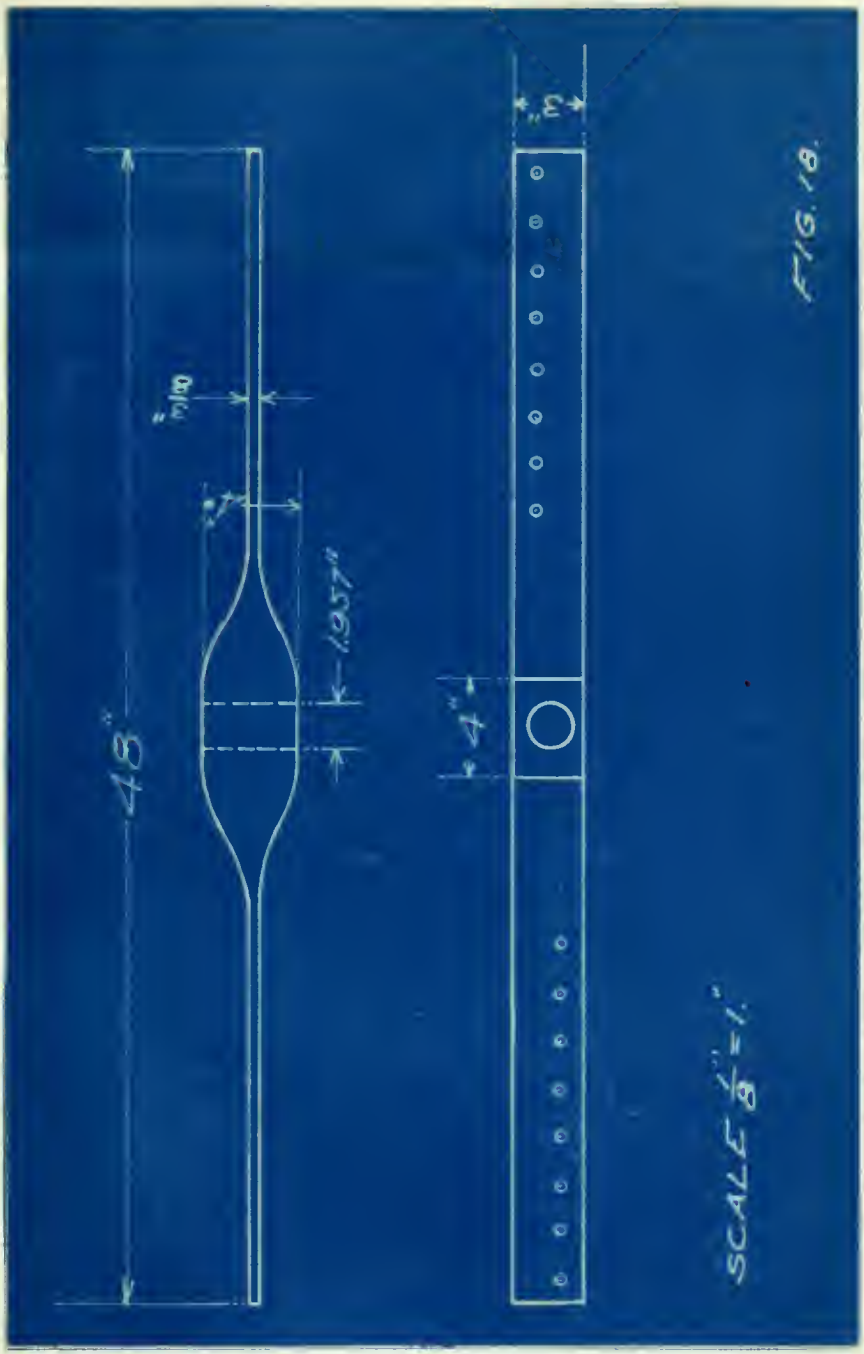


FIG. 18.

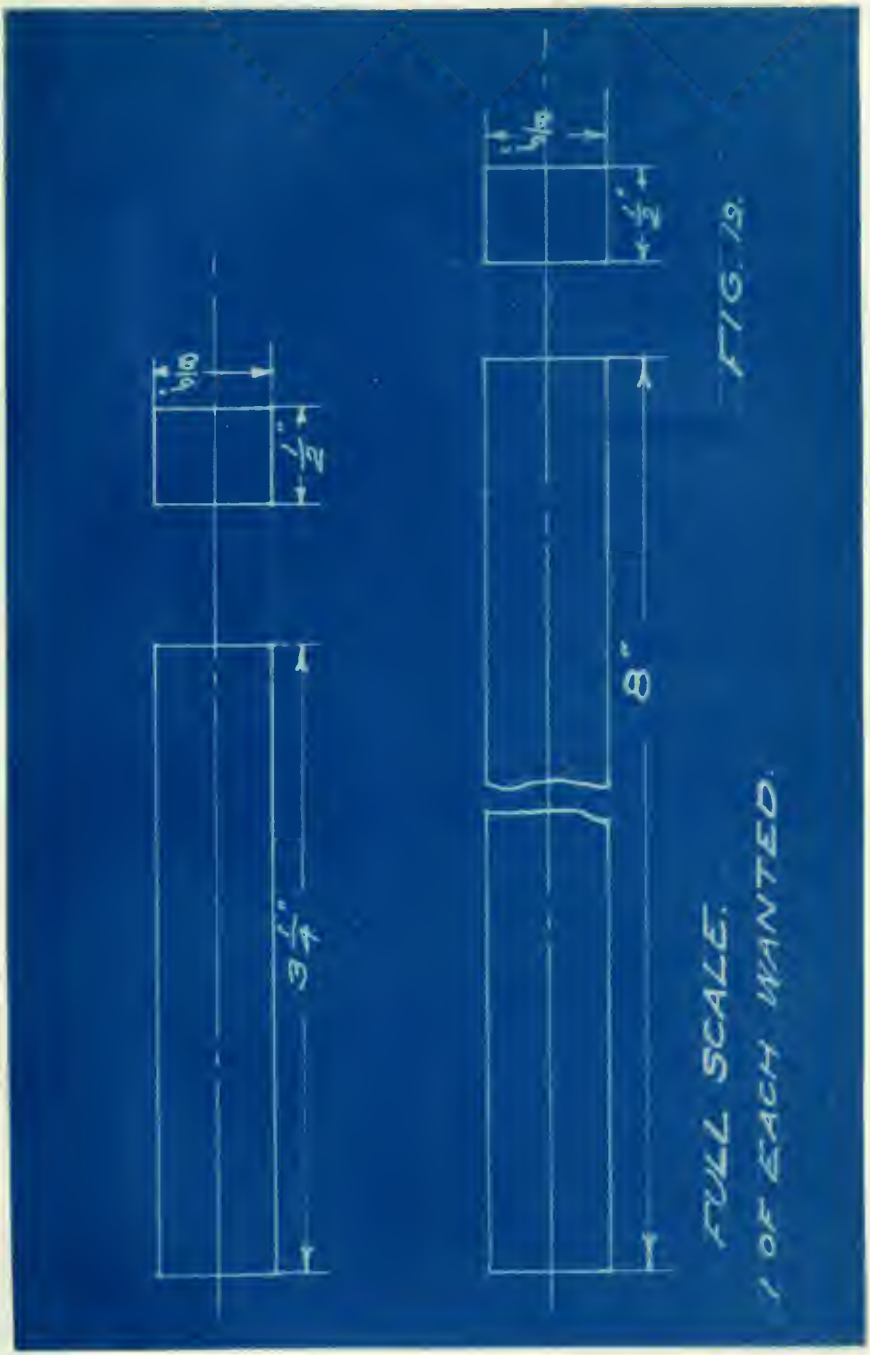
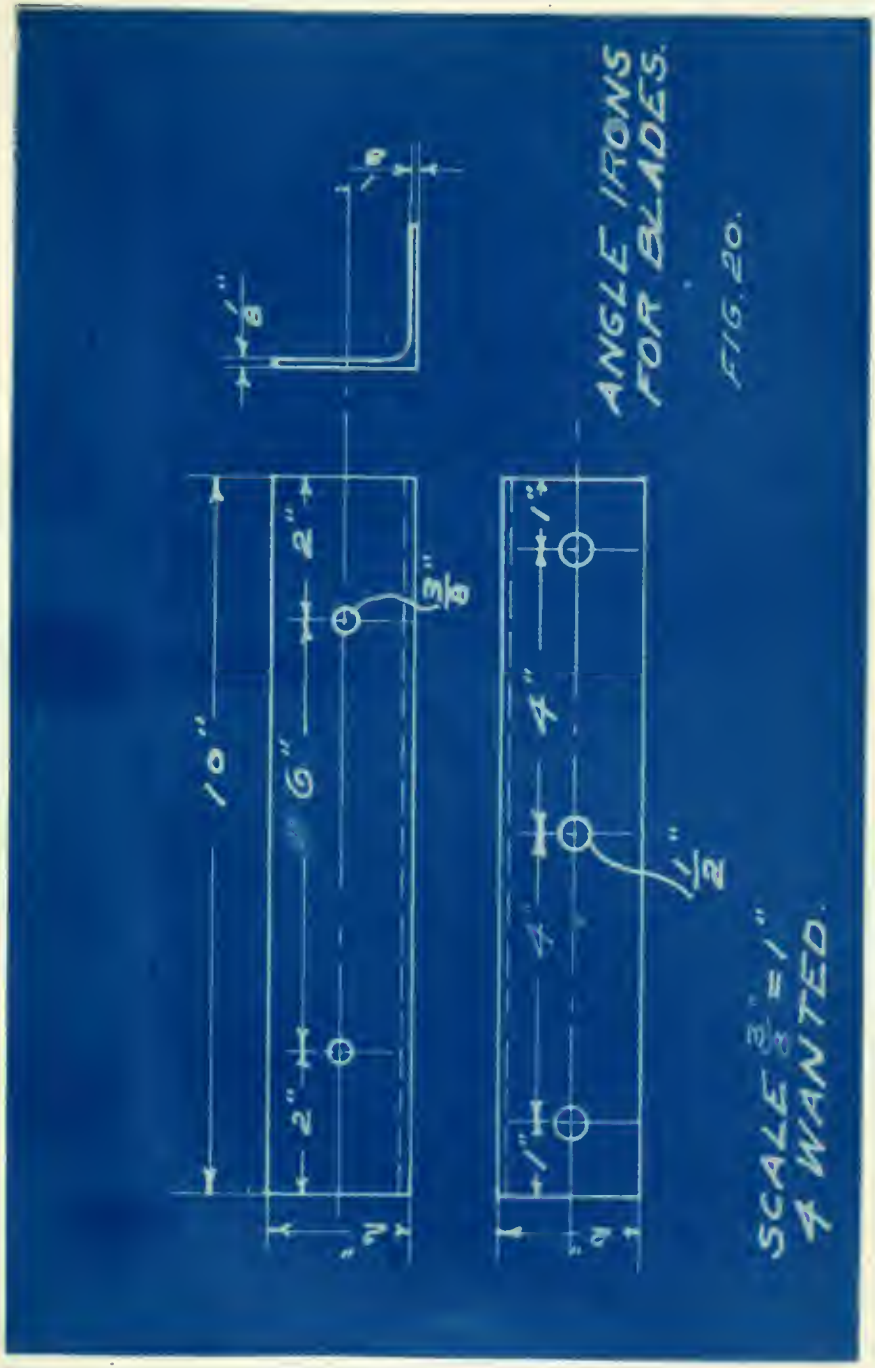


FIG. 19.


ANGLE IRONS
FOR BLADES.
FIG. 20.
SCALE 3/8" = 1"
IF WANTED.
10"
6"
2"
2"
3/8"
1"
4"
1"
1/2"
1"
2"
1"
1"

The keys were designed in this manner. The horsepower curve of the engine, see figure 23, page 46, shows a maximum of 40.5 H.P. at 1600 R.P.M., the engine running free. With the engine loaded down to a maximum with the fan brake, the speed will be reduced considerably. To find the fan speed with the blades at the greatest radius. Dimensions are approximately as follows: Assume standard conditions. Using the formula included in the S.A.E. report,

$40.5 = 6.9 \times 10^{-15} \times 2 \times n^3 \left[(11-0)(24^4 - 10^4) + 0XL \right]$

$n = 940$ R.P.M.

With Horseless Age equation,

$40.5 = \frac{n^3 \times 11 (24^{3.5} - 10^{3.5})}{68 \times 10^6}$

$n = 15.7$ R.P.S. $= 944$ R.P.M.

Assume R.P.M. of engine when loaded = 940 R.P.M.

To get size of key at flange.

Diameter of dynamometer shaft = 1.75"

$940 \times \pi \times \frac{1.75}{12} = 430$ ft/min.

$\frac{40.5 \times 33000}{430} = 3110$ lb. force acting.

$3110 = BL \times 2500$

$L = 3.25" =$ length of key.

$B = \frac{3180}{3.25 \times 2500} \rightarrow .392" = 7/16"$

$h = \frac{B}{2} = 7/32"$

Since the key will be subjected to inertia effects on the starting and stopping of the engine, it was decided to allow plenty for this.

The key was made 5/8" x 1/2", it being sunk 1/4" in the shaft and 1/4" in the coupling. Since both transmit the torque of the engine, it was decided to make the key for the arm of the same size.

The angle iron bolts were designed in the following manner. The following pieces were weighed. Weight of 3 bolts which hold angle iron to arm, total =.69lb.
Weight of 4 bolts which hold angle iron to plate, total = .32 lb.
Weight of plate = 2.83 lb.
Weight of angle iron = 1.24 lb.
Weight of angle iron = 1.23 lb.
Total weight = 6.31 lb.
To find diameter of bolt holding angle irons to plate, (four used). The only force will be centrifugal. It is

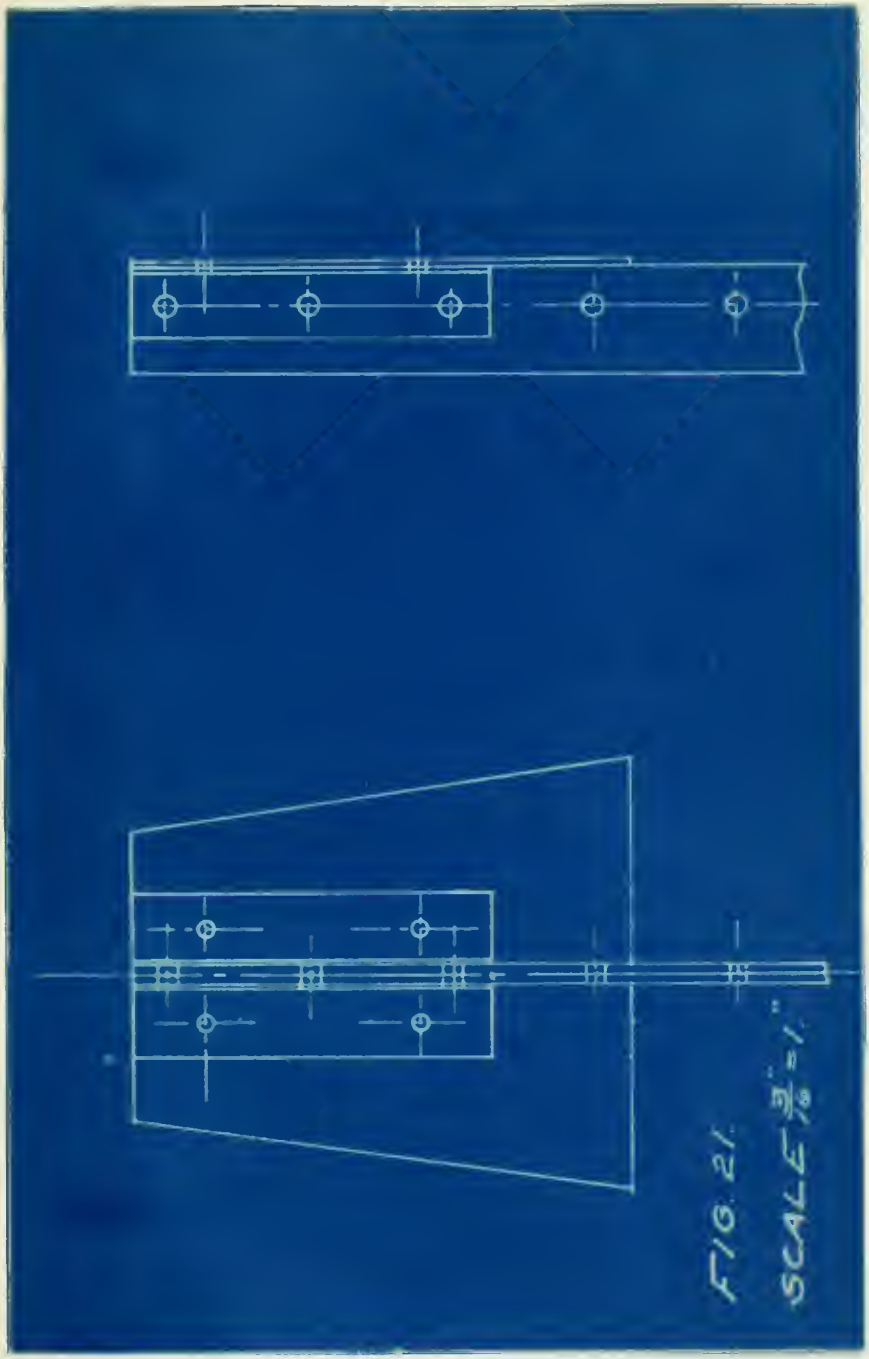


FIG. 21.
SCALE 3/16" = 1"

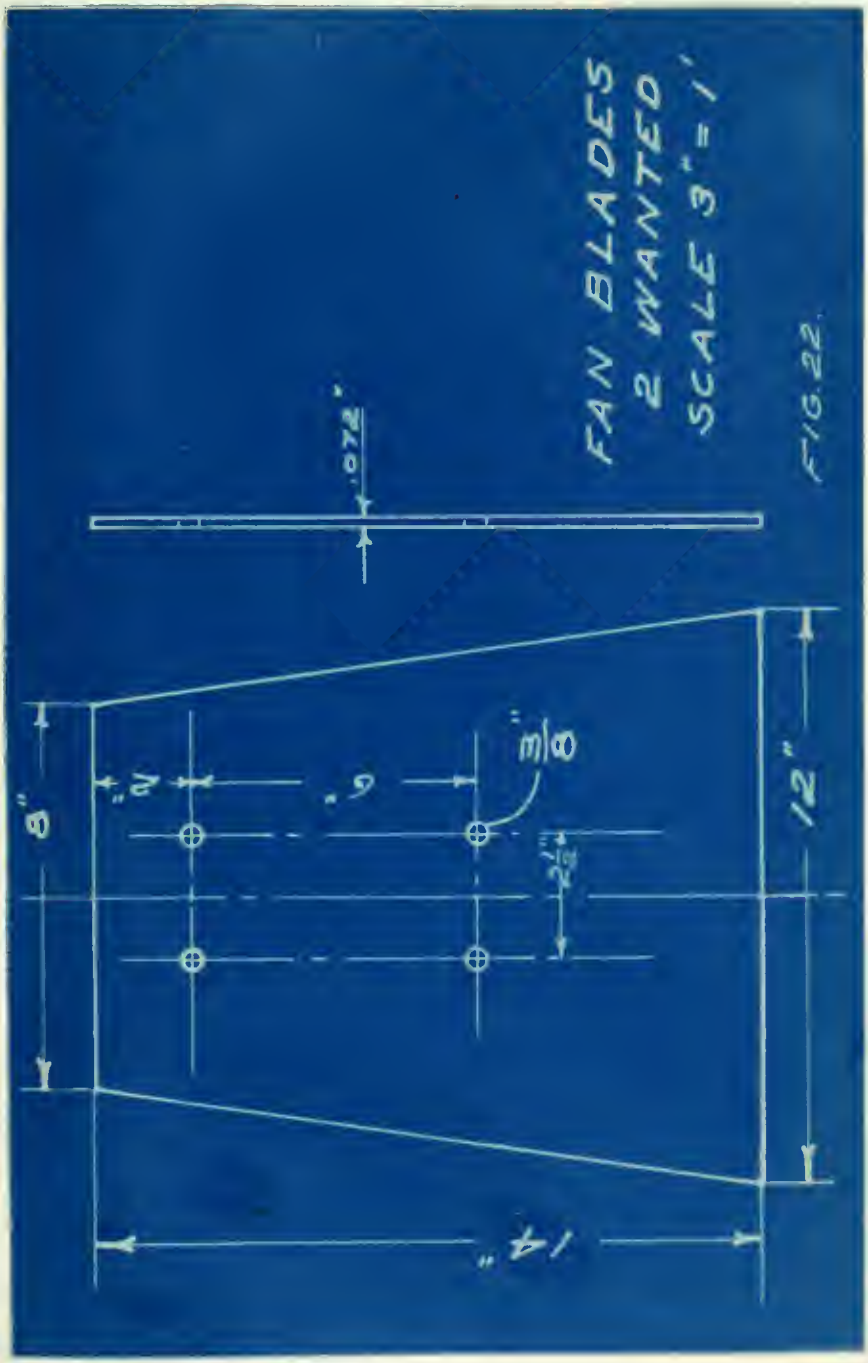


FIG. 22.

first necessary to find the speed of the engine with the blades in the innermost position. It is at this position that the engine will give maximum speed when developing maximum horsepower. We have

$$H.P. = \frac{n^3 w}{68 \times 10^6} \left(r_1^{3.5} - r_2^{3.5} \right)$$

Assume H.P. = 40.5

$w = 11"$

$r_2 = 14"$

$r_1 = 0"$

$$n^3 = \frac{40.5 \times 68 \times 10^6}{11 \left(14^{3.5} - 0\right)} = 24390$$

R.P.M. = 28.99 x 60 = 1738 R.P.M.

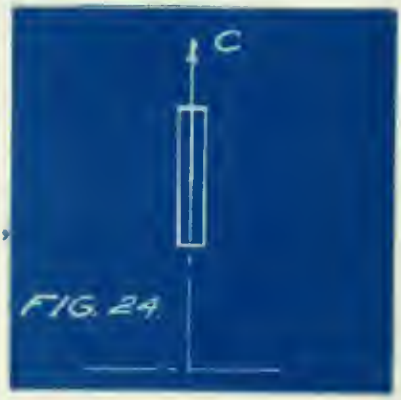
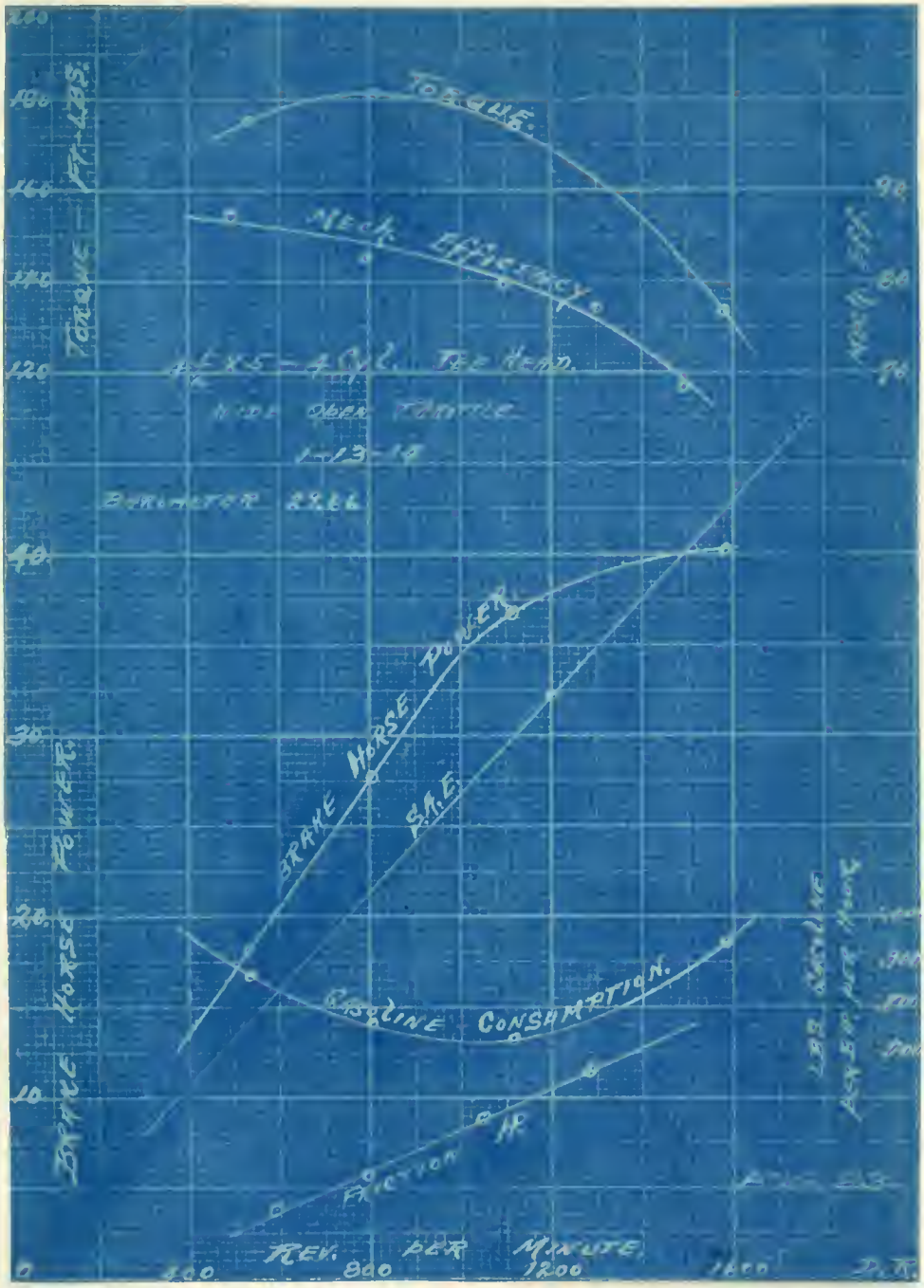


FIG. 24

This cannot be true for the engine will not develop 40.5 H.P. at 1738 R.P.M. However, the calculations will be on the safe side. The forces are shown in figure 24 page 45.

$$F = \frac{m v^2}{r}$$

$$m = \frac{2.83}{32.2} = .0878 \text{ lb.}$$


TORQUE.
Mech. Efficiency
4 Cyl. F-Head.
Side Oper. Valve
1-13-19
Barometer 29.66
BRAKE HORSE POWER
S.A.E.
Gasoline Consumption.
Friction H.P.
REV. PER MINUTE.
400
800
1200
1600
TORQUE - Ft-Lbs.
BRAKE HORSE POWER
Mech. Eff.
LBS. GASOLINE PER B.H.P. PER HOUR

$$v = \frac{1738 \times 2\pi \times 7}{60 \times 12} \doteq 106' / \text{sec.}$$

$$F = \frac{.0878 \times 106^2 \times 12}{7} = 1692 \text{ lb. force}$$

The bolts are in single shear.

$$1692 = 4\left(\frac{\pi}{4} d^2\right) \times \frac{45000}{8}$$

$$d = .31'' = \frac{3}{8}''$$

To design bolts to hold the angle irons to the arm. The fewer the number of bolts used the better, for each time the position of the plate on the arm is changed, the bolts must all be taken out and reset. Assume three bolts. There will be two forces acting, one centrifugal, and the other, the wind pressure on the plate. The resultant force is the one to be used.

This resultant force is the one to be used. This resultant force will tend to shear off the bolts. The two points at which the bolts would be most liable to shear is either at 1600 R.P.M. or at 940 R.P.M.

For 940 R.P.M. the centrifugal force is

$$F = \frac{mv^2}{r}$$

$$M = \frac{6.31}{32.2} = .196$$

$$V = \frac{940}{60} \times 2\pi \times \frac{17}{12} = 139.4'/\text{sec.}$$

$$F = \frac{196 \times 139.4^2 \times 12}{17} = 2686\#$$

The pressure acting is as follows:

At 940 R.P.M. the torque of the engine is

$$T = \frac{40.5 \times 33000 \times 12}{940 \times 2 \times 17} = 159.9\#$$

$$P = \frac{159.9 \times 12}{17} = 113\#$$

$$R = \sqrt{2686^2 + 113^2} = 2689\#$$

$$2689 = \frac{45000}{8} \times 3 \left(\frac{\pi}{4} d^2\right)$$

$$d = .450" = 1/2"$$

The highest speed at which the engine will deliver 40.5 H.P. is 1600 R.P.M. For 1600 R.P.M. The centrifugal force is

$$M = .196$$

$$V = \frac{1600}{60} \times 2\pi \times \frac{7}{12} = 97.6'/\text{sec.}$$

$$F = \frac{.196 \times 97.6^2 \times 12}{7} = 3204\#$$

The pressure

$$T = \frac{40.5 \times 33000 \times 12}{1600 \times 2\pi \times 7} = 228\#$$

$$P = \frac{228 \times 12}{7} = 392\#$$

$R = \sqrt{3204^2 + 392^2} = 3226\#$

$3226 = \frac{45000x\ 3(\frac{\pi}{4}d^2)}{3}$

$d \neq .49'' = 1/2''$

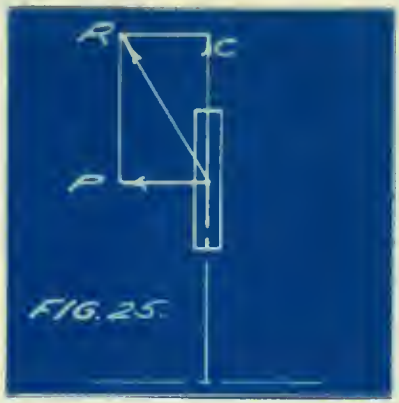


FIG. 25.

A means of registering the reaction force was next designed. It was decided to use a spring balance suspended from an A frame, see fig.26, page 50, and hooked at the bottom to the end of the dynamometer that went down when the reaction came into play. A pronged piece fitted over the end of the dynamometer arm and was tapped out in its other end, see fig.27. A hook, threaded on the straight end, screwed into this piece and hooked onto the balance. The upper hook was threaded and run through a hole in the top of the A frame, again see fig.26. It was bolted on top with a nut and washer. A calibration curve of the spring balance is shown in fig.28. The calibration curve was obtained in this manner.

As the engine was placed in the frame, its cen-

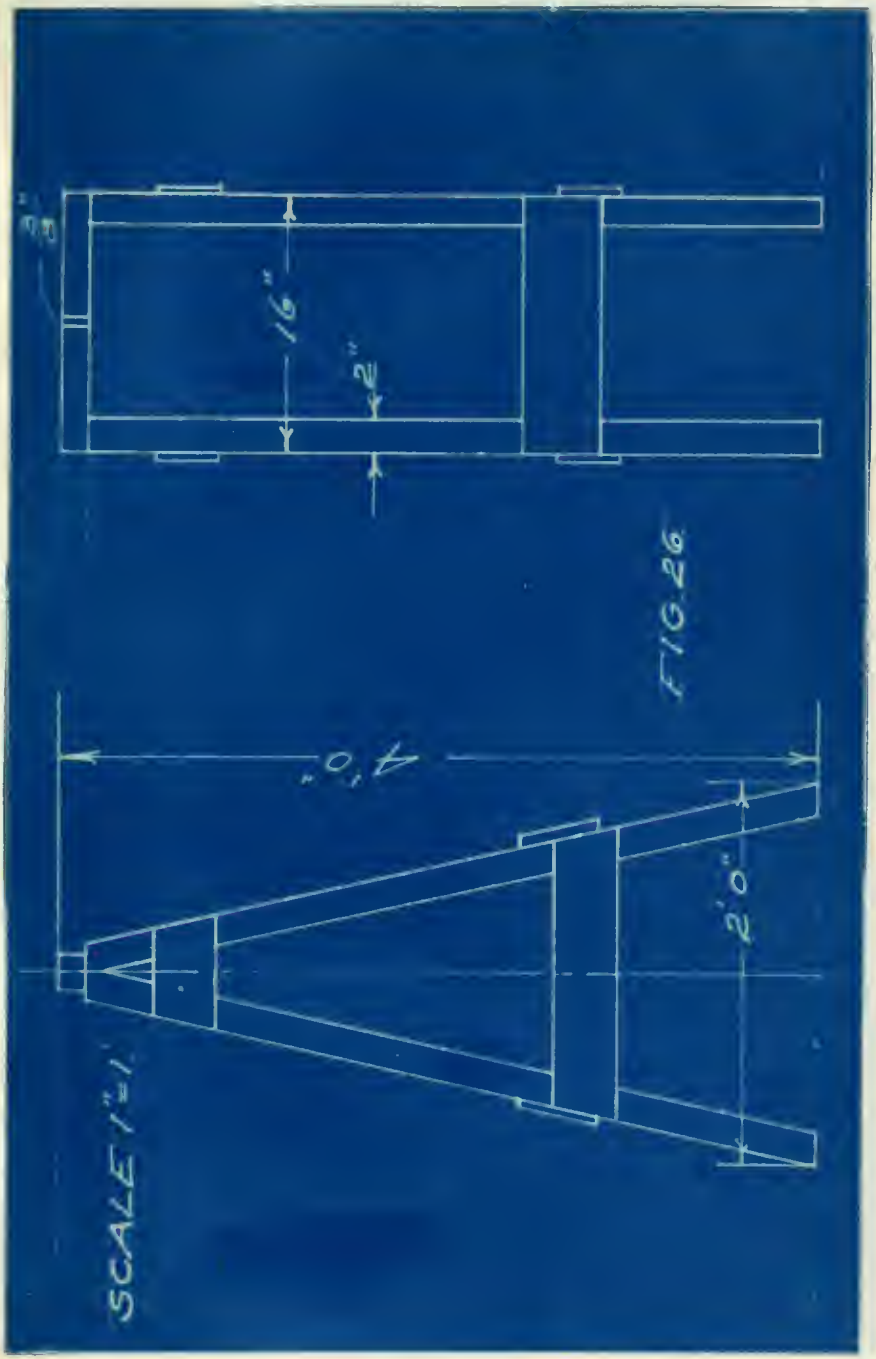


FIG. 26

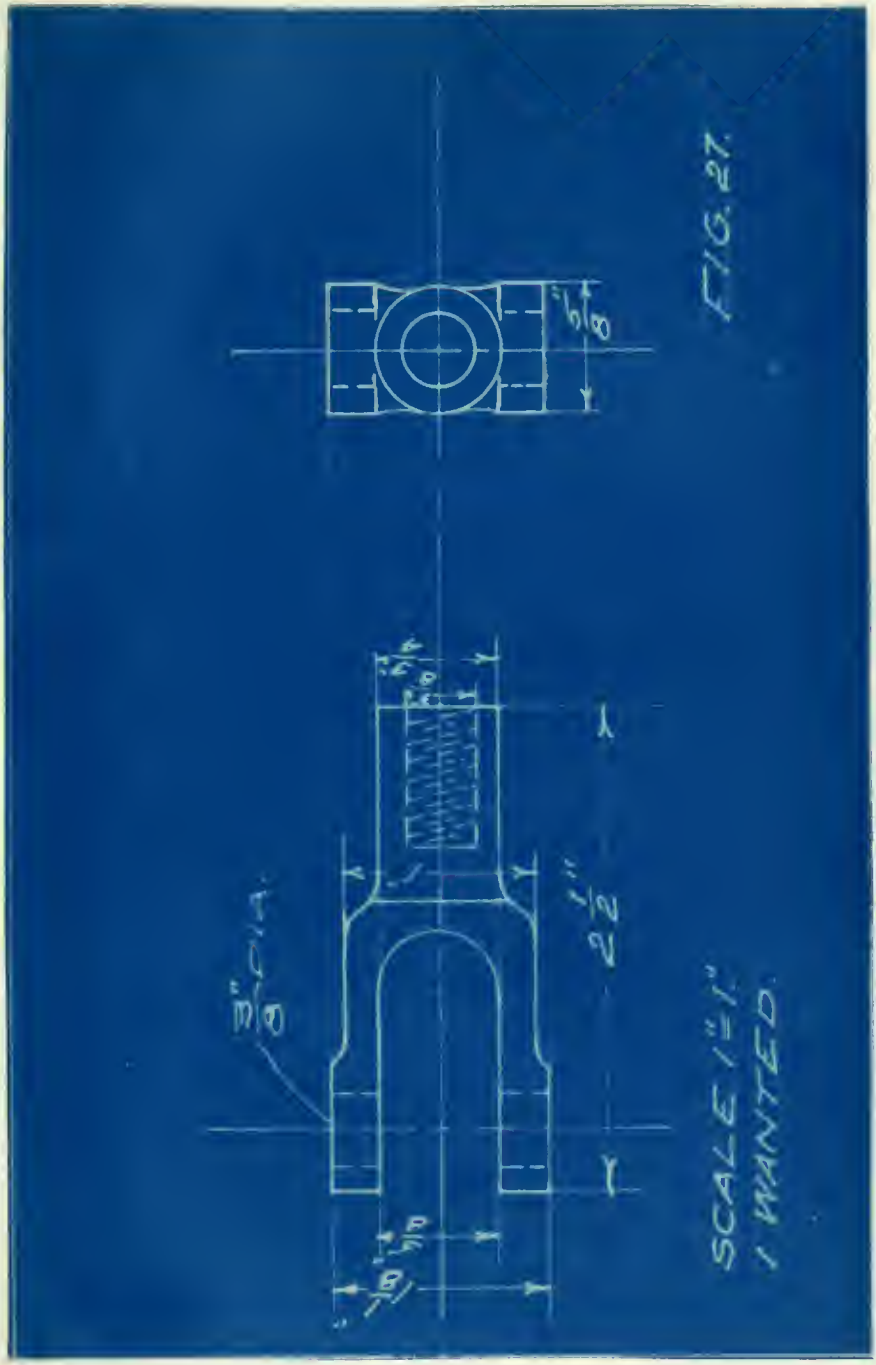
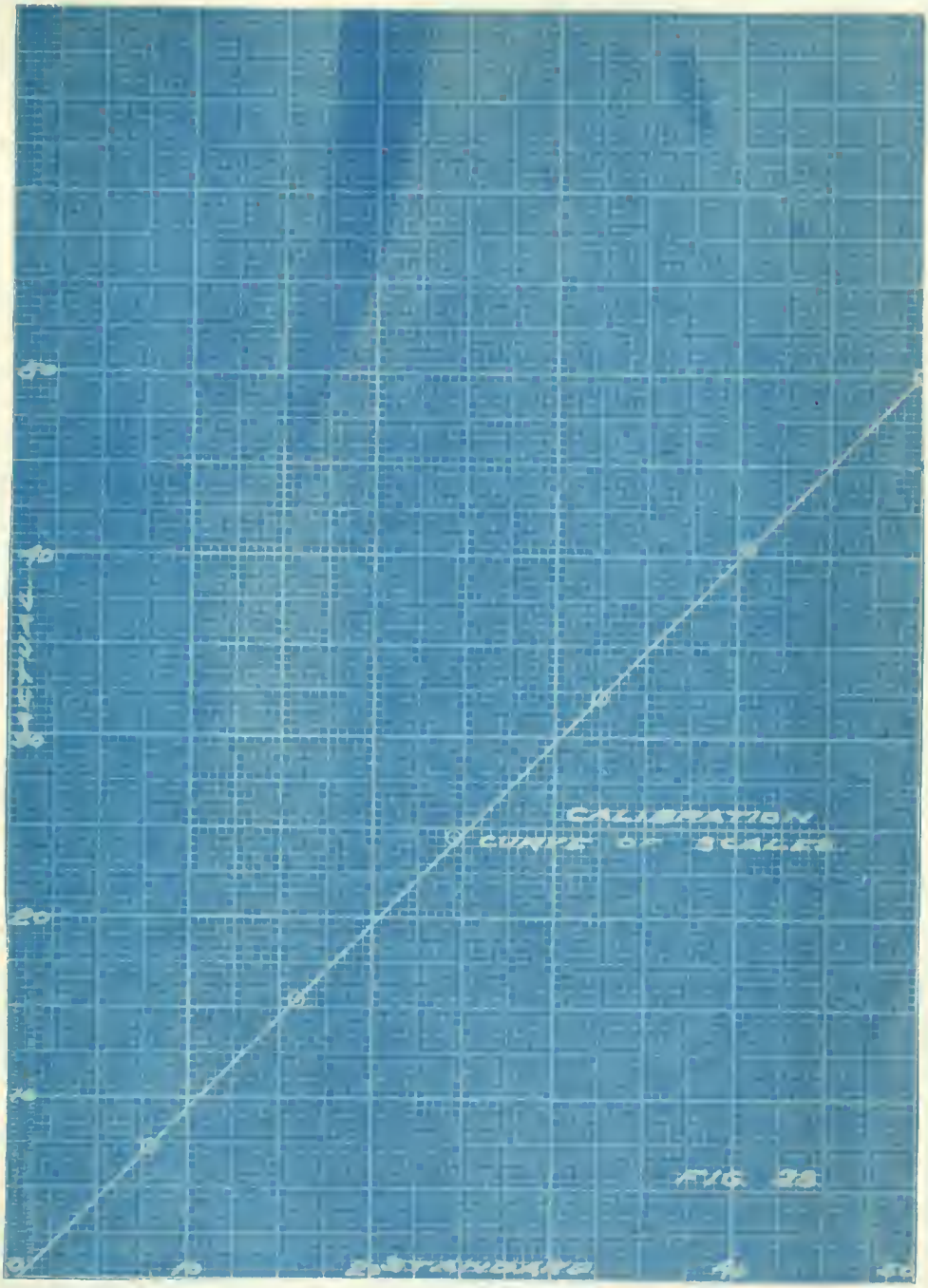


FIG. 27.


CALIBRATION
CURVE OF SCALES
FIG. 28

ter of gravity was higher than the center of the reaction dynamometer bearings. As a result, when the engine started to tip, due to reaction, it would fall until it hit the stops. Fig.29 explains this. At the start, there is only the reaction force acting. When the engine begins to tip, however, there is a component acting due to its weight, and this component grows larger as the engine tips more and more. There are two ways to remedy this: One is to calibrate the spring balance, when it is in position hooked on to the dynamometer arm. When the various standard weights are placed on the balance, the engine weights are placed on the balance, the engine weight component will be automatically taken care of, since it will come in on the weight of the arm as it

Fig. 29.

pushes down. The other way is to lower the center of gravity of the engine until it is below the dynamometer bearings. Then the weight component of the engine will not enter, and the weight registered on the scale will be the true force if the engine is balanced horizontally by means of another arm extending out the other side of the reaction dynamometer. Iron washers, placed on this arm, balanced the engine in this direction, see fig.1. The radiator brought in complications for it was placed off center of the reaction dynamometer bearings. As a result it made one side of the engine heavy so that it tended to stay on those stops all the time, and this force varied, depending upon the amount of water in the radiator.

The center of gravity of the engine was lowered in this manner. Underneath the reaction dynamometer on a line through its center of balance, was suspended a steel rod which was threaded. A large nut

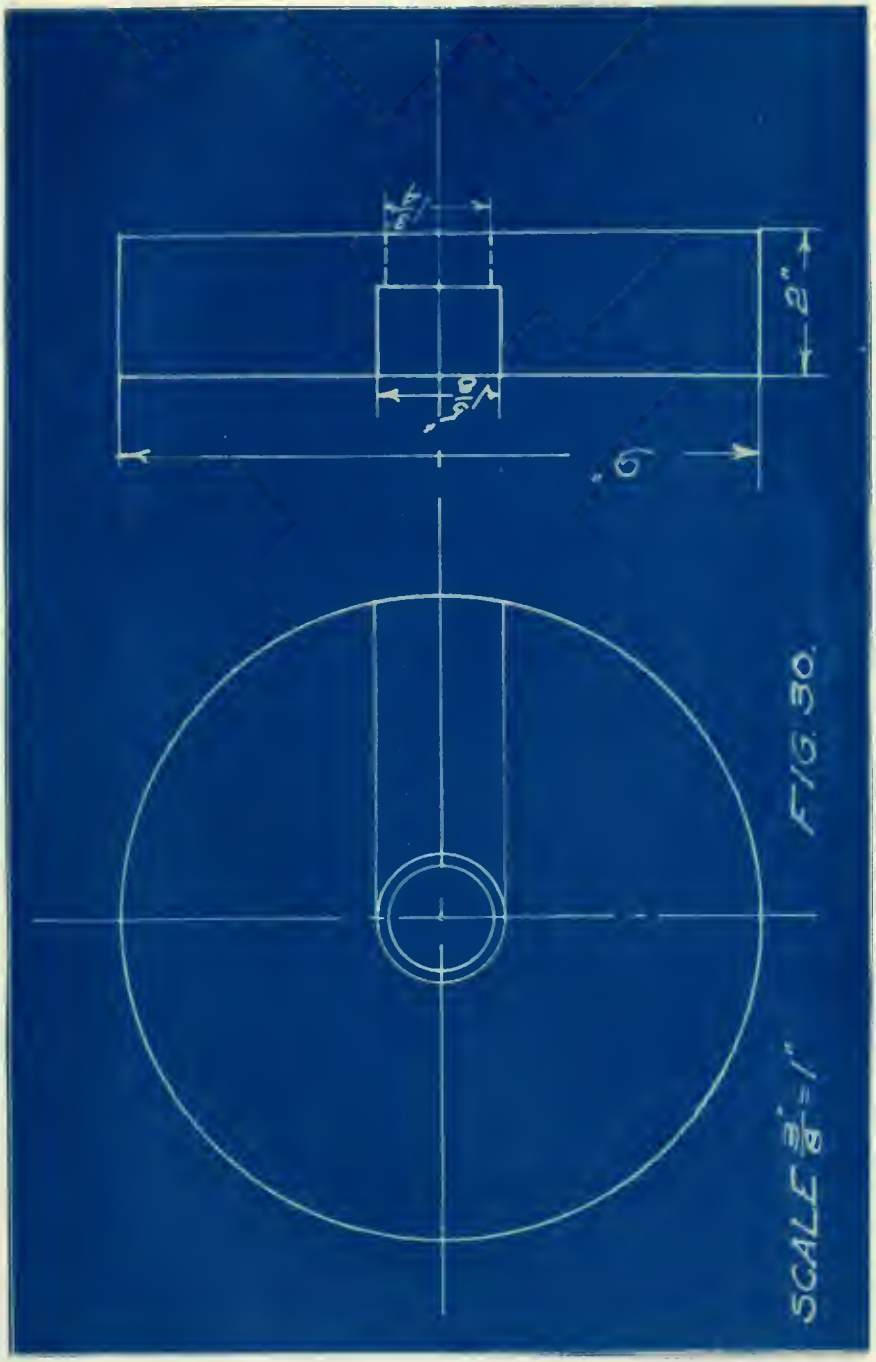
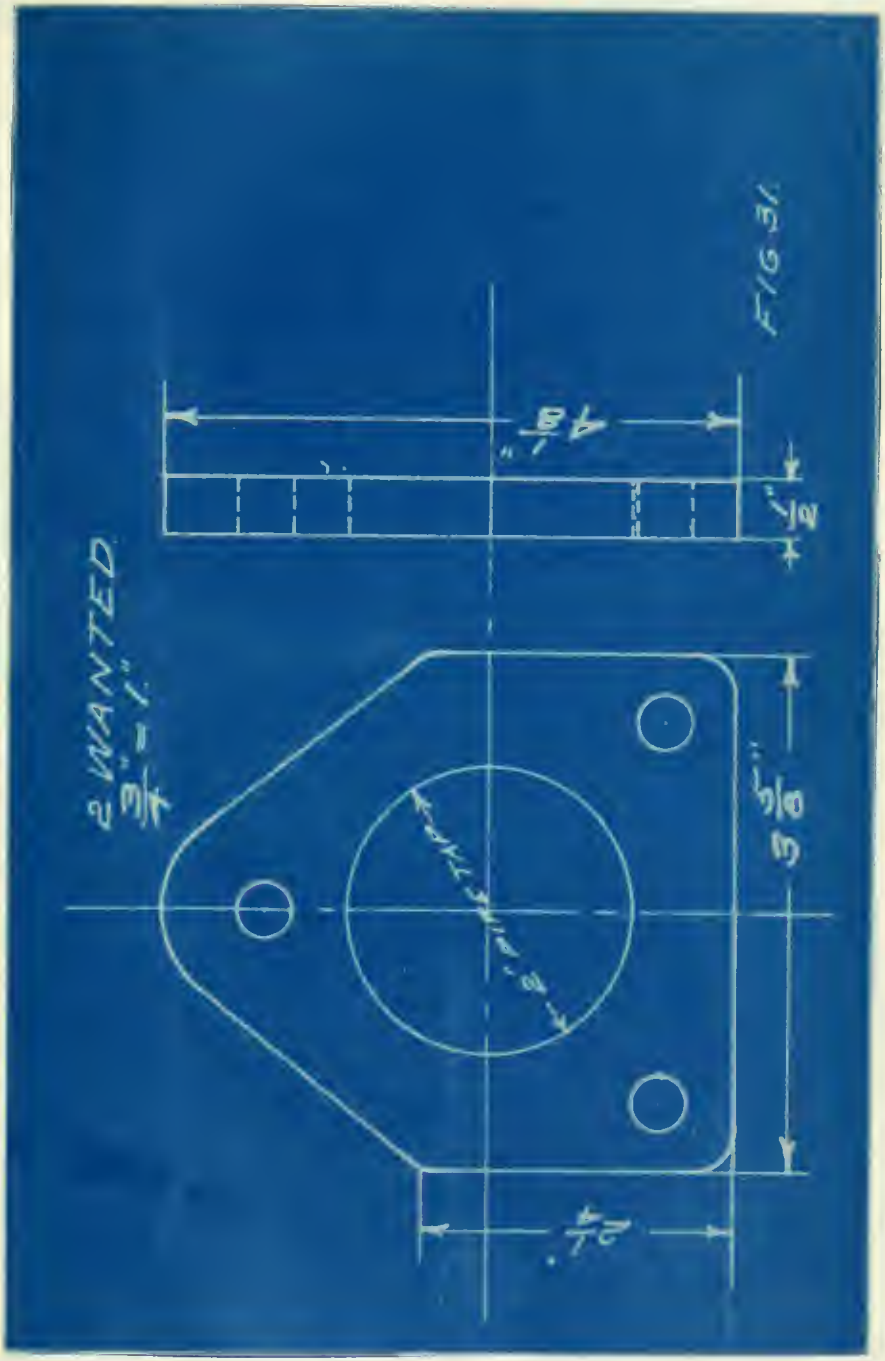


FIG. 30.


2 WANTED
FIG 31

screwed on this rod. Weights were furnished to place around this rod and on top of the nut. With all of the weights on, however, it was found that the engine was still top heavy. It was attempted to get the nut off the rod, but in setting up the engine, the rod was lowered so that there was not enough clearance between the floor and the bottom of the rod to get the nut off. At the top, the bolts holding the rod to the frame were too long in the same manner.

It was recognized that a heavy weitht at the bottom of this rod would give a maximum of counter-balancing force, against the top-heaviness of the engine. A piece of cast iron was machined up as shown in fig.30 and screwed on to the bottom of the rod. A similar piece, slotted out, and the original weights helped to lower the center of gravity of the engine, below the center line of the bearings until it was nearly in balance. The scale was then calibrated while it was fastened to the dynamometer arm and the scale was set to read correctly at 50#.

With the exhaust pipes of the engine extending straight up in the air, a reaction would be formed from the escaping exhaust gases. This was done away with by sending the gases out on a line parellel to the centerline of the crankshaft of the engine. The exhaust pipes were attached to the engine by flanges, shown in fig. 31, page 56 . In this position no reaction was formed.

There is another reaction formed. When the carbureter sucks up gasoline there is an area of one or two square inches in the bottom of the carburetor, wherein the pressure is lowered. This will cause a reaction, but it is so small that it can be neglected.

The gasoline was fed from an elevated tank by gravity into the engine. A flexible rubber tubing connection made any interference with the recording of the proper reaction a negligible quantity.

The friction of the ball bearings was so small that it was neglected.

A cage was designed, see fig. 32, page 60, to cover the fan dynamometer. The cage was built by M.H. Hickey, Chicago, but it came so late that there was not time left to run a test with it in use. The cage had the upper circular part hinged so that the top could be taken off, if necessary.

The materials which were ordered outside were obtained from the following concerns. The steel plate for the blades came from C.G. Stevens, Chicago. The bolts holding the plates to the angle irons and the angle irons to the arm came from the Beckley Ralston Co., Chicago. The ball bearings came from the Hess- Bright Co., Chicago. The base came from Joseph Tracey, New York. The cage, as was stated above, came from M.H. Hickey, Chicago.

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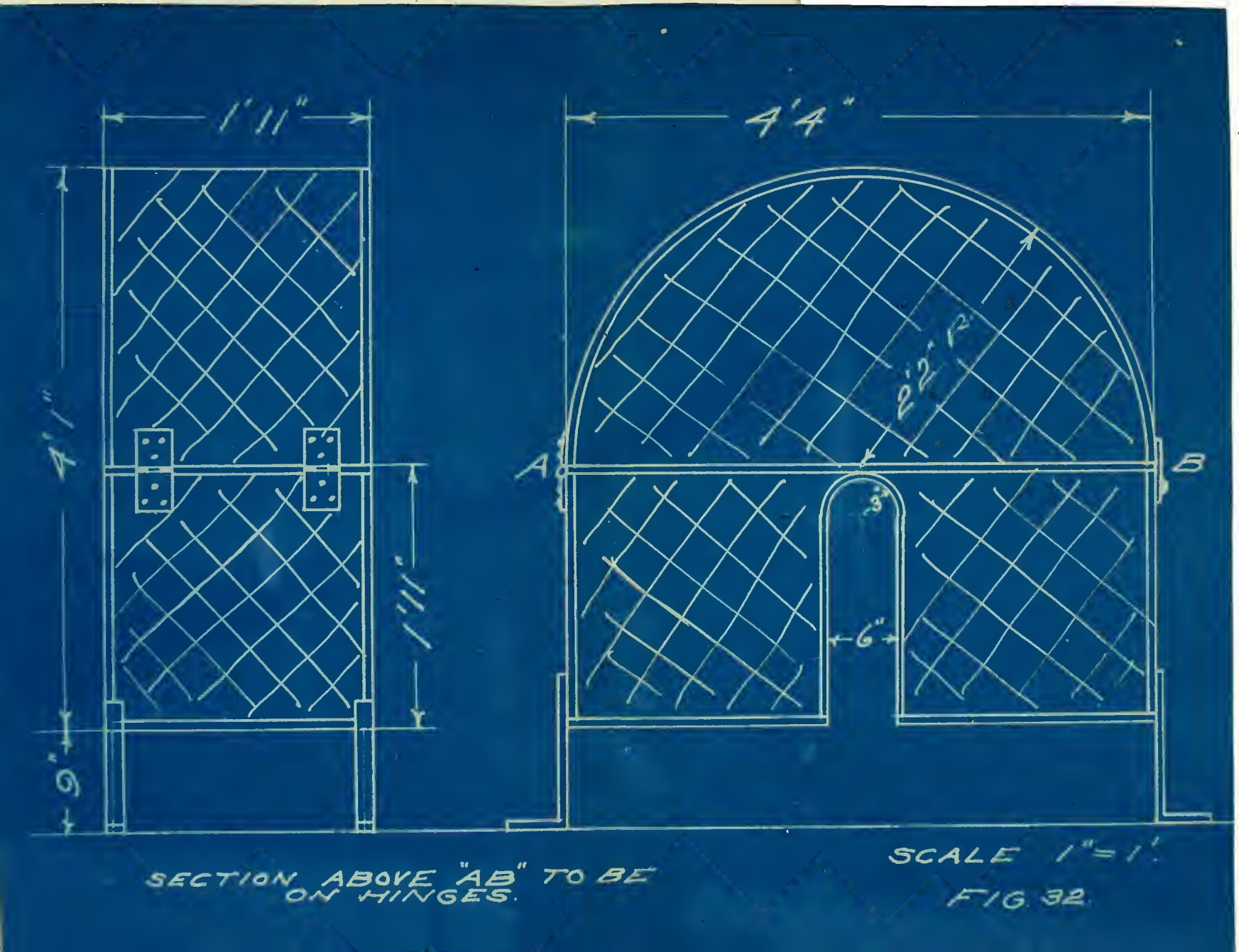

SECTION ABOVE "AB" TO BE ON HINGES.

SCALE 1"=1'

FIG. 32.

TESTING AND RESULTS.

oOo

Time was so limited, when the dynamometer was assembled, that but one test was made. It was run on the afternoon of May 11th,1916. The cage as yet had not come, so the test was made without it. The scale was first calibrated as was explained above, and the engine was balanced horizontally with the radiator full of water. Before each run the radiator was filled, so that the balance could be maintained. The relative humidity was measured by a wet and dry bulb thermometer, see fig1, placed in the room wherein the test was made. The air pressure was measured by a barometer wherein the zero of the scale was adjusted and a correction was made for the temperature. The temperature was registered by a theremometer placed in an adjoining room so that it registered the temperature of the air untouched by the warm gasoline vapor. The speed was read with a calibrated tachometer,made by

the Industrial Instrument Co., Foxboro, Mass. The run was made with the windows and doors of the room opened so that several currents of air were blowing across the apparatus. The relative humidity and the barometric pressure were each read once for the test. Five runs in all were made, one for each of the five positions of the plates on the arms. One man run the engine and adjusted the power, by means of the throttle, so that certain readings were obtained on the scale. The second man read the speed of the engine by means of the tachometer. The third man took data. Various loads were obtained on the scale, starting at a mimimum and rising to a maximum. For each run, the position of the plates was changed. The data as taken is shown on Fig. 33, page 65, The carburetor used in this test was smaller than the one used in obtaining the curves in fig. 23, so that forty horsepower was not developed.

The calculations for horsepower in this test are very simple. The following will serve as a sample. By the reaction dynamometer

$$H.P. = \frac{W\ N}{2000}$$

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TEST OF FAN DYNAMOMETER MAY 11, 1916 — TEST CONDUCTED BY: ROESCH, WILLETT, ECHLIN.

AREA OF BLADES — 140 SQ. IN.

RELATIVE HUMIDITY — 43%. BAROMETER — $29.52\left[\left(1-\{78-32\}.000010\right)\right] = 29.38$

LENGTH OF ARM IN INCHES	TEMPERATURE											
24.0	72°	LOAD	14.0	14.0	20.0	25.0	26.0	28.0	36.0	40.0	41.0	48.0	50.0
		R.P.M.	420	420	510	530	560	580	650	780	710	770	770
		H.P.	2.94	2.94	5.10	6.63	7.28	8.12	11.70	15.60	14.57	18.48	19.27
22.0	72°	LOAD	15.2	20.0	25.0	29.5	37.0						
		R.P.M.	550	650	730	860	800						
		H.P.	4.18	6.50	9.12	12.68	14.80						
20.0	74°	LOAD	7.5	15.0	19.5	25.0	34.0	39.0	44.0				
		R.P.M.	520	650	750	830	900	950	1020				
		H.P.	2.14	4.88	7.32	10.38	15.30	18.52	22.44				
18.0	74°	LOAD	10.0	15.0	21.0	24.0	29.0	35.0	39.0	40.5			
		R.P.M.	670	740	880	980	1070	1100	1150	1190			
		H.P.	3.35	5.55	9.24	11.77	15.52	19.25	22.42	24.10			
16.0	74°	LOAD	7.0	11.0	13.0	15.5	18.0	22.0	28.0				
		R.P.M.	640	750	850	930	1040	1220	1220				
		H.P.	2.24	4.13	5.52	7.21	9.36	16.48	17.09				

FIG. 33.

For run one, reading one

$$W = 14.0 \text{ lb.}$$

$$N = 420 \text{ R.P.M.}$$

$$H.P. = \frac{14.0 \times 420}{2000}$$

$$= 2.94 \text{ H.P.}$$

The correction for temperature was made in the following equation.

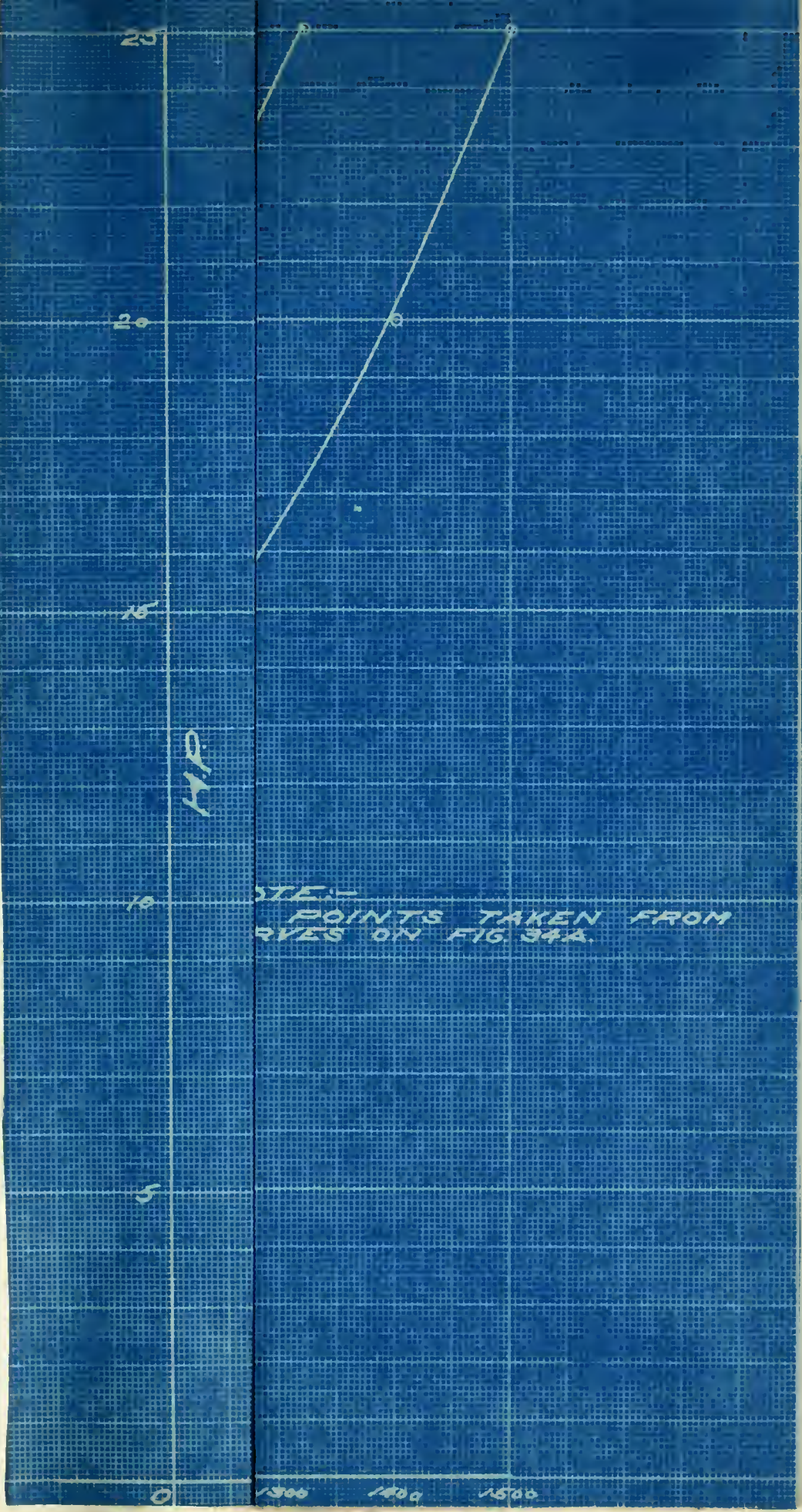
$$\text{Correct reading} = 29.52 \left[\left(1 - [78-32] \times .000101 \right) \right]$$

$$= 29.38'' \text{ Hg.}$$

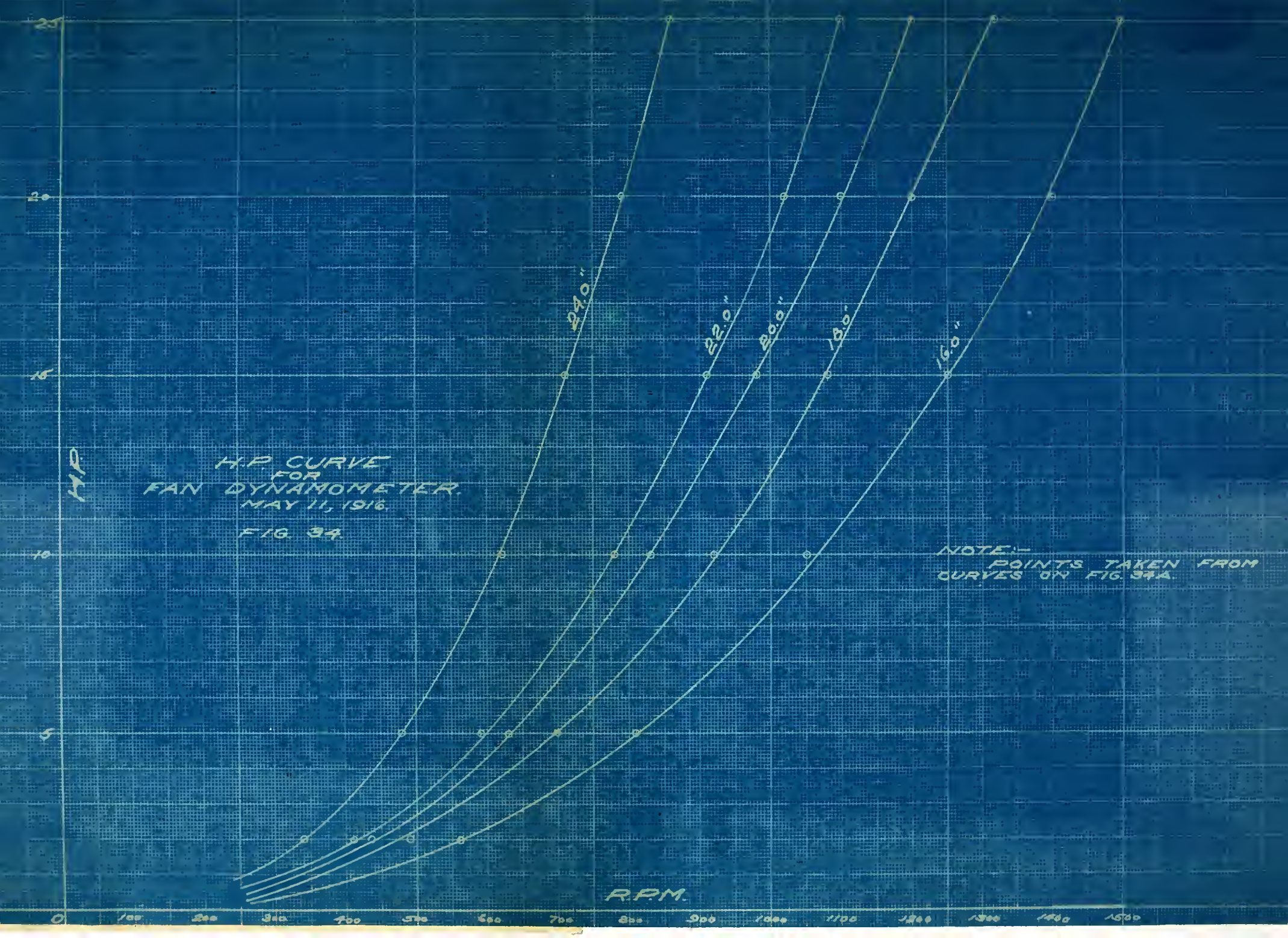
With the value for horsepower obtained in this manner, the curves in fig.34, page 67, were plotted. These curves show the general characteristics of the fan dynamometer. The longer the arm and the higher the speed, the greater will be the horsepower that can be absorbed. Fig.34 A, page 68, shows these same curves plotted on logarithmic paper.

The curves are approximately straight lines although the points for high speeds do not check up well. Fig.37 A, page 76, shows the relation between the speed and the radius of the arms. Fig.

Page 67 at back.



NOTE:—
POINTS TAKEN FROM
CURVES ON FIG. 34A.



H.P. CURVE
FOR
FAN DYNAMOMETER.
MAY 11, 1916.

FIG. 34.

NOTE:—
POINTS TAKEN FROM
CURVES ON FIG. 34A.

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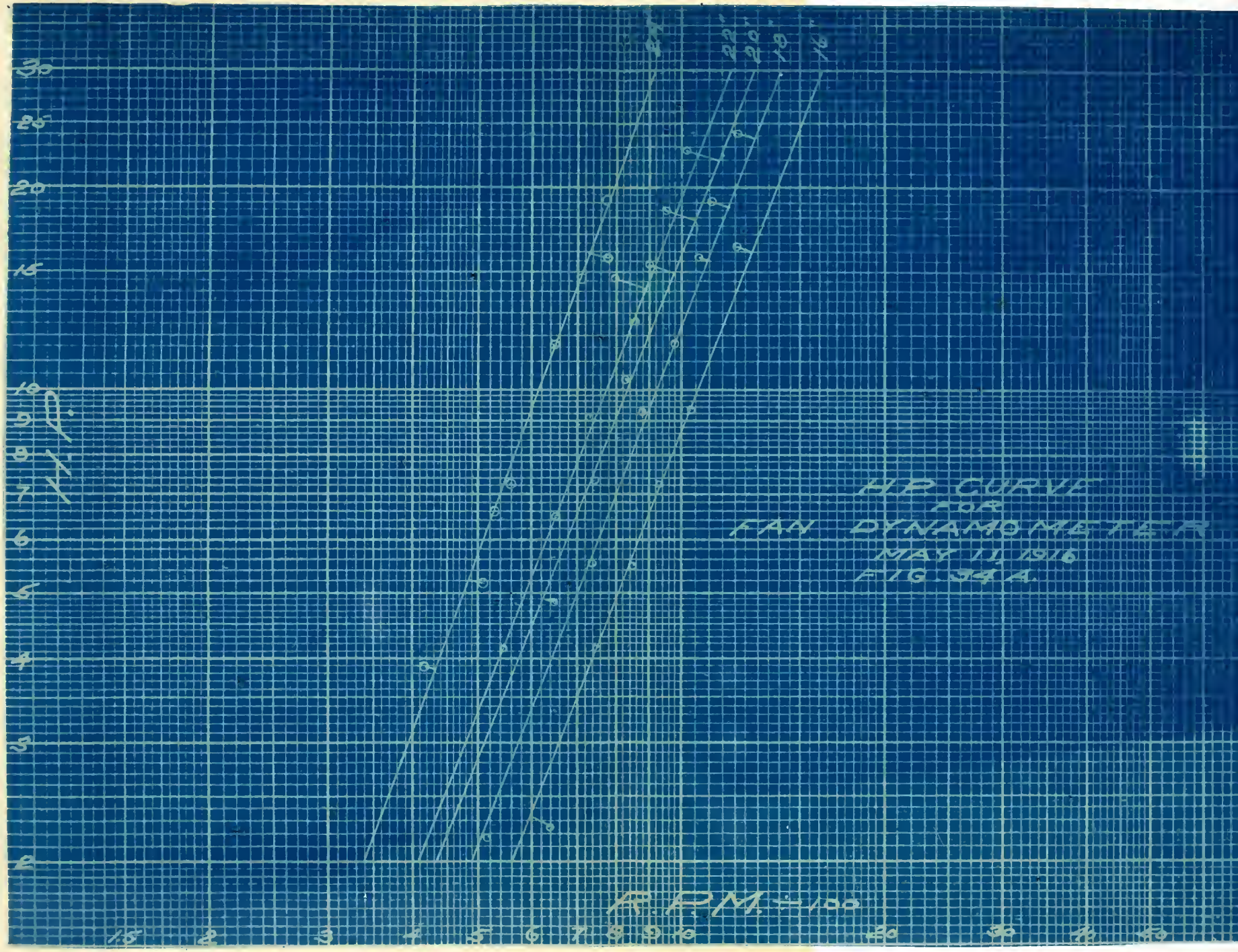

E.M.F.
R.P.M. = 100
E.M.F. CURVE
FOR
FAN DYNAMOMETER
MAY 11 1916
FIG. 34 A.

37, page 75, shows these curves plotted on logarithmic paper. These curves check out very well as straight lines. Fig.38 A, page 80, shows the relation between horsepower and the radius of the arms. These curves are approximately straight lines. Fig.38, page 79, shows these curves plotted on logarithmic paper. The curves check out as straight lines fairly well except for the outermost position of the blades.

An equation based on these results can now very easily be obtained. The following shows the value of n, obtained from the logarithmic curves.

Values of n.

Run	H.P.-$\frac{R.P.M.}{100}$	$\frac{R.P.M.}{100}$-rad.	H.P.-rad.
1	2.71	.57	1.47
2	2.48	.57	1.59
3	2.46	.60	1.82
4	2.56	.60	1.70
5	2.52	.58	1.90
Ave.	2.546	.584	1.696

The radius spoken of is the radius of the center of pressure of the plate. The dimensions of the plate are shown in fig.22. The center of pressure of the plate alone is

$$C = \frac{(7 \times 8 \times 14) + 2\ (14 \times 2 \times .5 \times 3.66)}{(2 \times 2 \times .5 \times 14) + (8 \times 14)}$$

$$= 6.53"$$

For the outermost position of the blades the distance of the center of pressure from the center of the shaft is

$$D = 24 - 14 + 6.53 = 16.53"$$

The same distance for any other position of the blades can be found in the same manner.

An experimental equation can now be evolved. We have

$$H.P. = c\left(\frac{N}{100}\right)^{2.546}$$

$$R.P.M. = c\ (r)^{.584}$$

$$H.P. = c\ (r)^{1.696}$$

$$H.P. = c\ (r)^{1.696}$$

Then $$H.P. = K(r)^{1.696}\left(\frac{N}{100}\right)^{2.546}$$

The data from several of the runs made was used in this equation and an average value for K obtained.

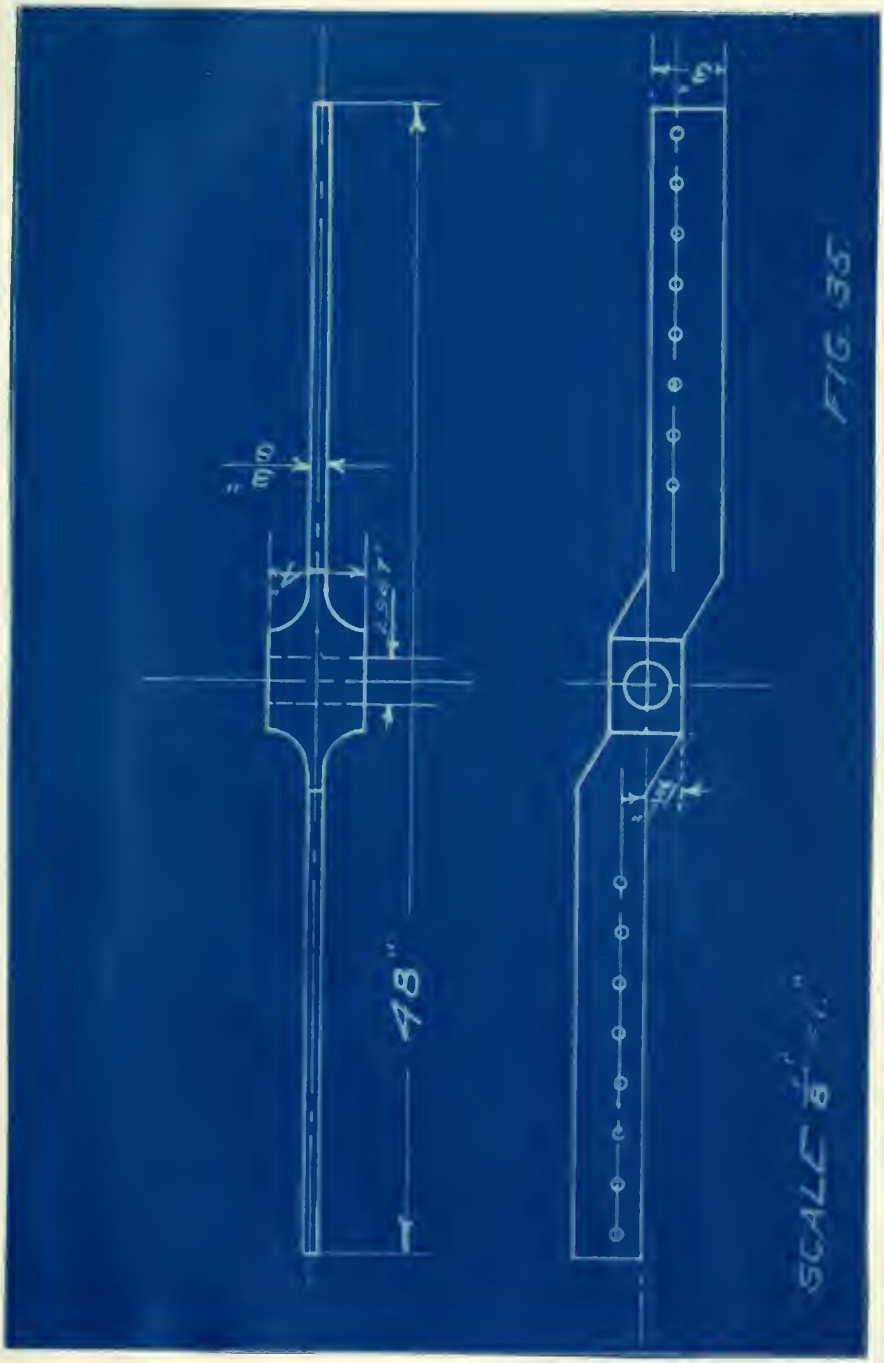


FIG. 35

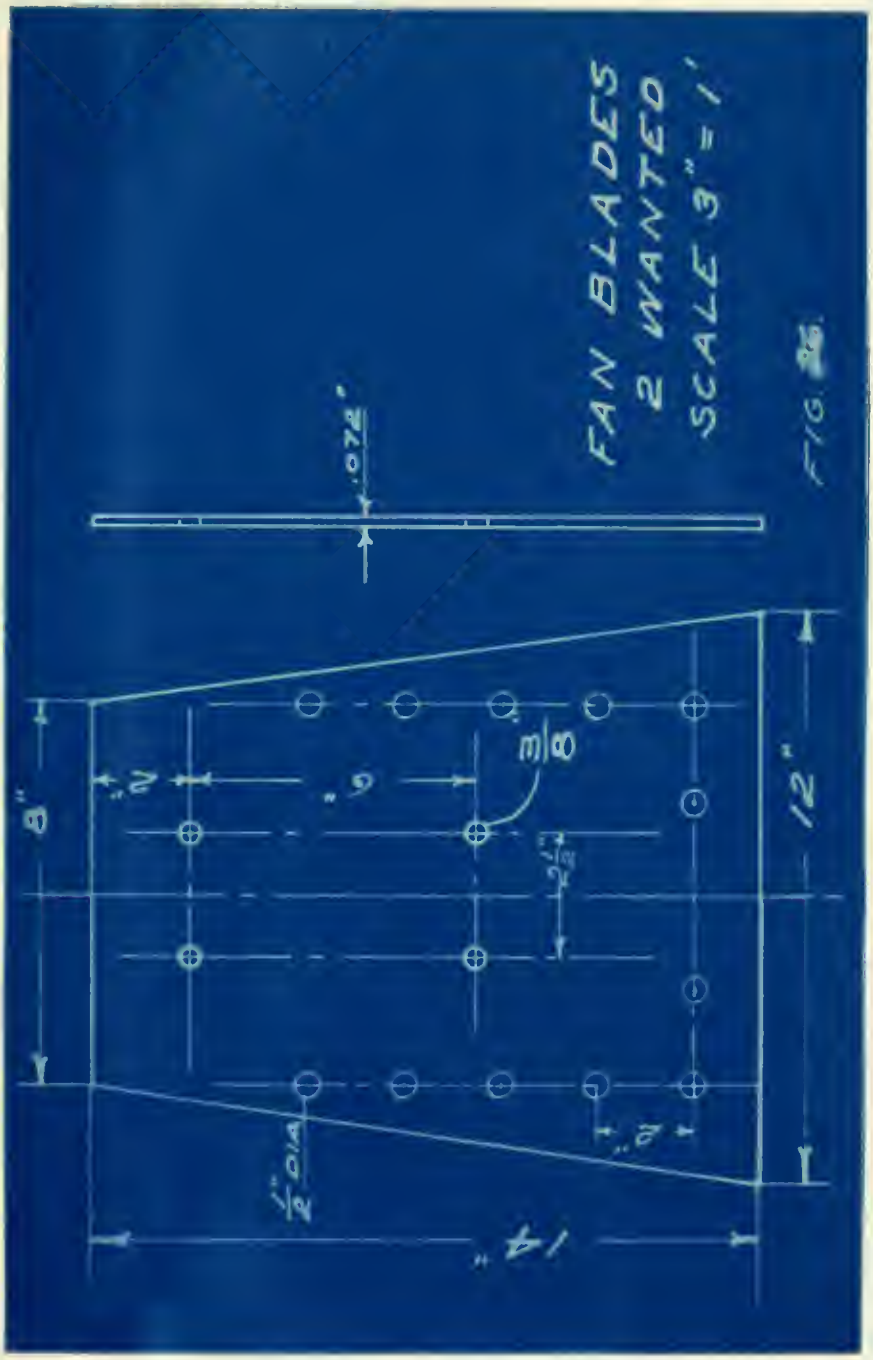


FIG.

$$K = .000838$$

$$\text{Then H.P.} = .000838\ (r)^{1.69}\left(\frac{N}{100}\right)^{2.546}$$

This equation can be expanded into a more general and a more approximate form. Its is known that the horsepower depends upon the density of the air that surrounds the fan.

Let γ = density of the air.

= .0807 lb. cu.ft. at 32 F. 29.92" Hg. and 0% relative humidity.

Then for any temperature, pressure and relative humidity

$$\gamma = \left[.0807 \times \frac{492}{T} \times \frac{B}{29.92} + R\ V\right]$$

$$= \left[\frac{1.329}{T}\ B + R\ V\right]$$

where B = observed barometric pressure

T = absolute observed temperature

R = percent relative humidity

V = density of the water vapor in the air in pounds per cubic foot.

The horsepower also depends on the area of the blades. Approximating the powers of r and $\left(\frac{N}{100}\right)$

$$\text{H.P.} = C \times A \times (r)^{1.7} \times \left(\frac{N}{100}\right)^{2.5} \left[\frac{1.329}{T}\ B + R\ V\right]$$

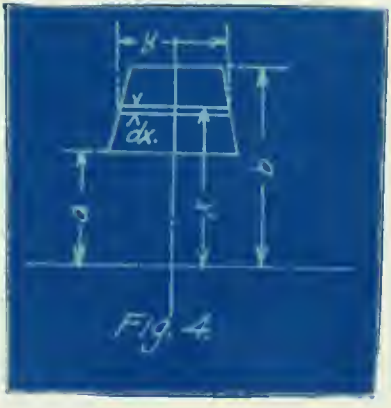
Again solving for C

$$H.P. = .000085\ A\ (r)^{1.7}\left(\frac{N}{100}\right)^{2.5}\left(\frac{1.329\ B + R\ V}{T}\right)$$

where A = area of plate in sq. in.

A logical way of developing a horsepower equation for a fan dynamometer is this. When the fan dynamometer is revolving, the resistance on the fan blade is caused by a certain head of air, pressing on these blades.

Fig. 4, page 74, shows the dimensions. On a strip y dx there will be a certain pressure. This pressure can be found in the following manner and with it can be obtained an equation for the horsepower developed.



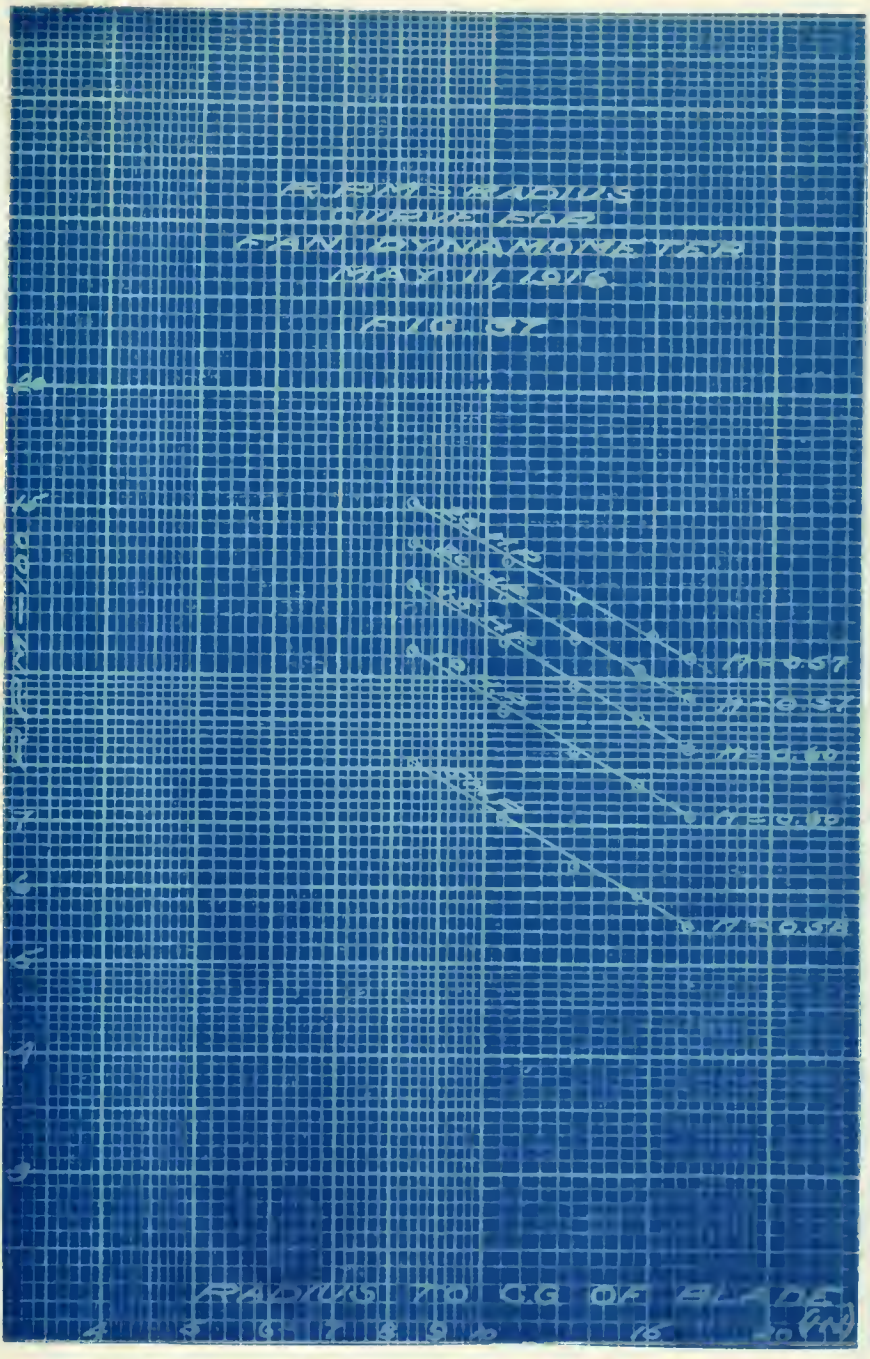
Let v = speed of the strip dx in ft. /sec.

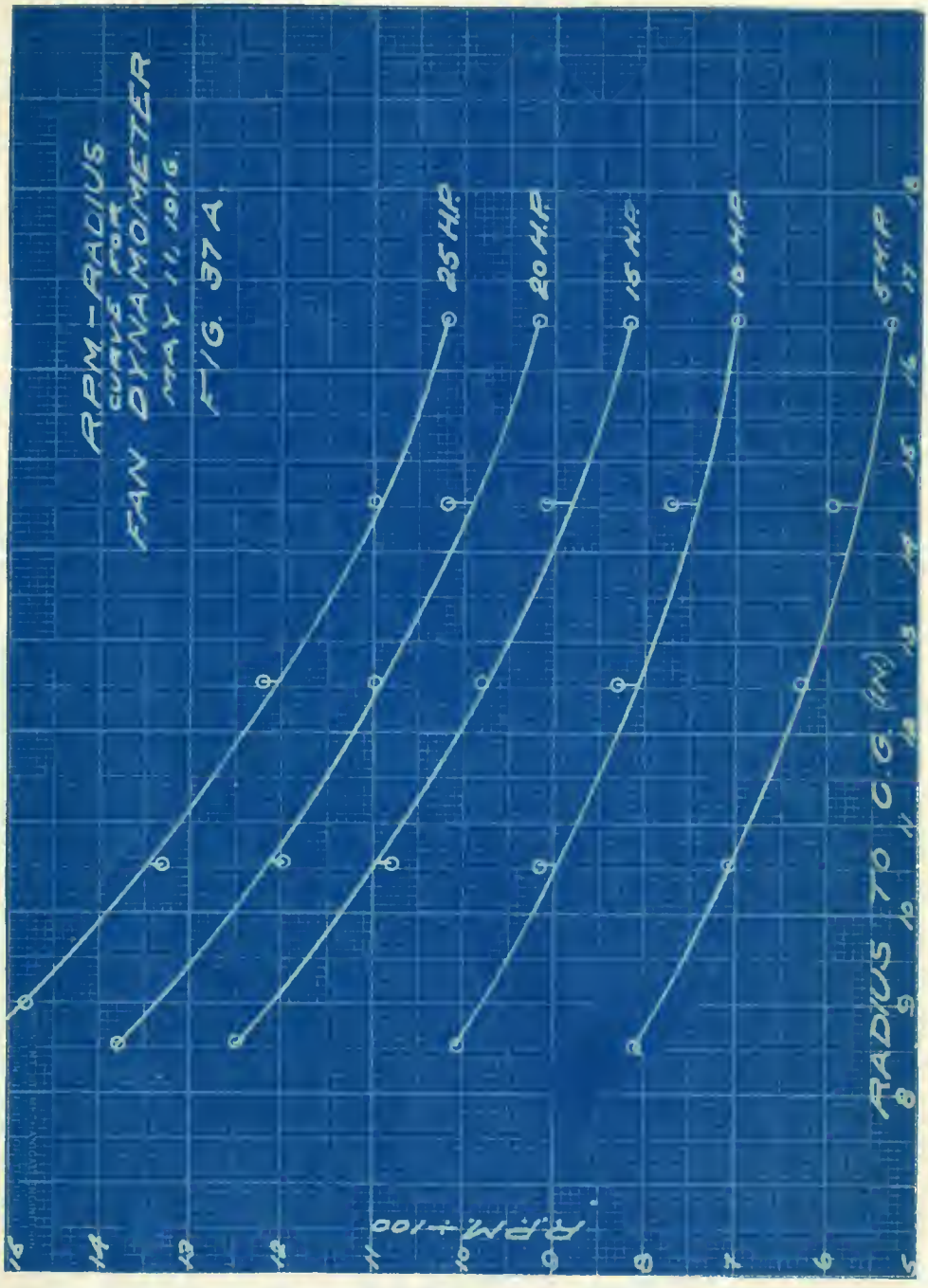
h = feet head of air on the strip dx.

g = 32.16

y = density of the air as shown above.

p = pressure of the air per unit area of blade surface.


FLIGHT RADIUS
WITH 50%
STANDARD PAYLOAD
MAX. ALT. 10K
FIG. 37


R.P.M.-RADIUS
CURVES FOR
FAN DYNAMOMETER
MAY 11, 1916.
FIG. 37 A.
25 H.P.
20 H.P.
15 H.P.
10 H.P.
5 H.P.
R.P.M. ÷ 100
RADIUS TO C.G. (IN.)

F = total pressure acting on the two plates.

N = R.P.M. of the fan dynamometer.

$n = \frac{R.P.M.}{1000}$

Other dimensions are as shown in fig.4.

$h = \frac{V^2}{2g}$

$P = H\gamma = \frac{v^2\gamma}{2g}$

The pressure acting on the area y dx

$= \frac{v^2\gamma}{2g} y \ dx$

$v = \frac{2\pi x N}{60}$ ft./ sec

$F = \frac{2 v^2 \gamma y}{2 g} dx$

$= \frac{8\pi^2 N^2 \gamma y}{7200 g} \int_a^b x^2 \, dx$

$H.P. = \frac{2\pi x F N}{33000}$

$= \frac{16\pi^3 N^3 \gamma y}{7200 g \times 33000} \int_a^b x^3 dx$

$= \frac{16\pi^3 \times 1000000000}{7200 \times 32.16 \times 33000} n^3 \gamma y \int_a^b x^3 dx$

$H.P. = 16.2 \, n^3 y \, (b^4 - a^4) \left[\frac{1.329 B + R V}{T}\right]$

Using this equation with several of the runs, it was found that it gave about one third of the

horsepower, as given by the reaction dynamometer. Such an equation should give accurate results, as it takes into account the barometric pressure, temperature, and relative humidity of the air, and uses no constants determined experimentally.

A change in design of the arm which would make it theoretically correct is shown in fig. 35, page 71. In the arm as built, see fig. 18, the plane of the blade would not pass through the centerline of the dynamometer shaft. To make this theoretically correct, an arm could be designed as shown in fig. 35. Fig. 36, page 72, shows a blade which could be used to determine whether the friction of the air had any influence on the horsepower developed, as there would be friction of the air through the holes.

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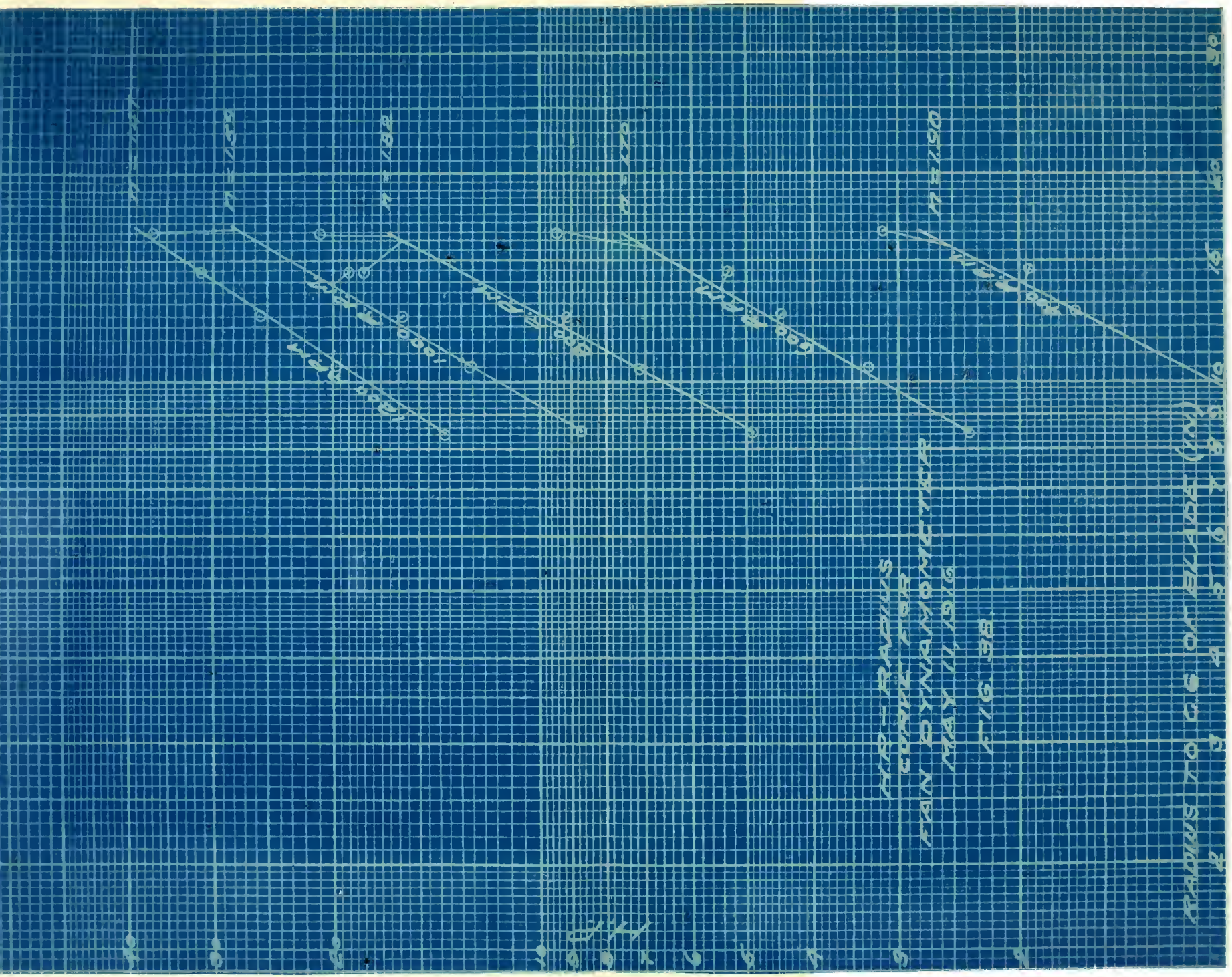

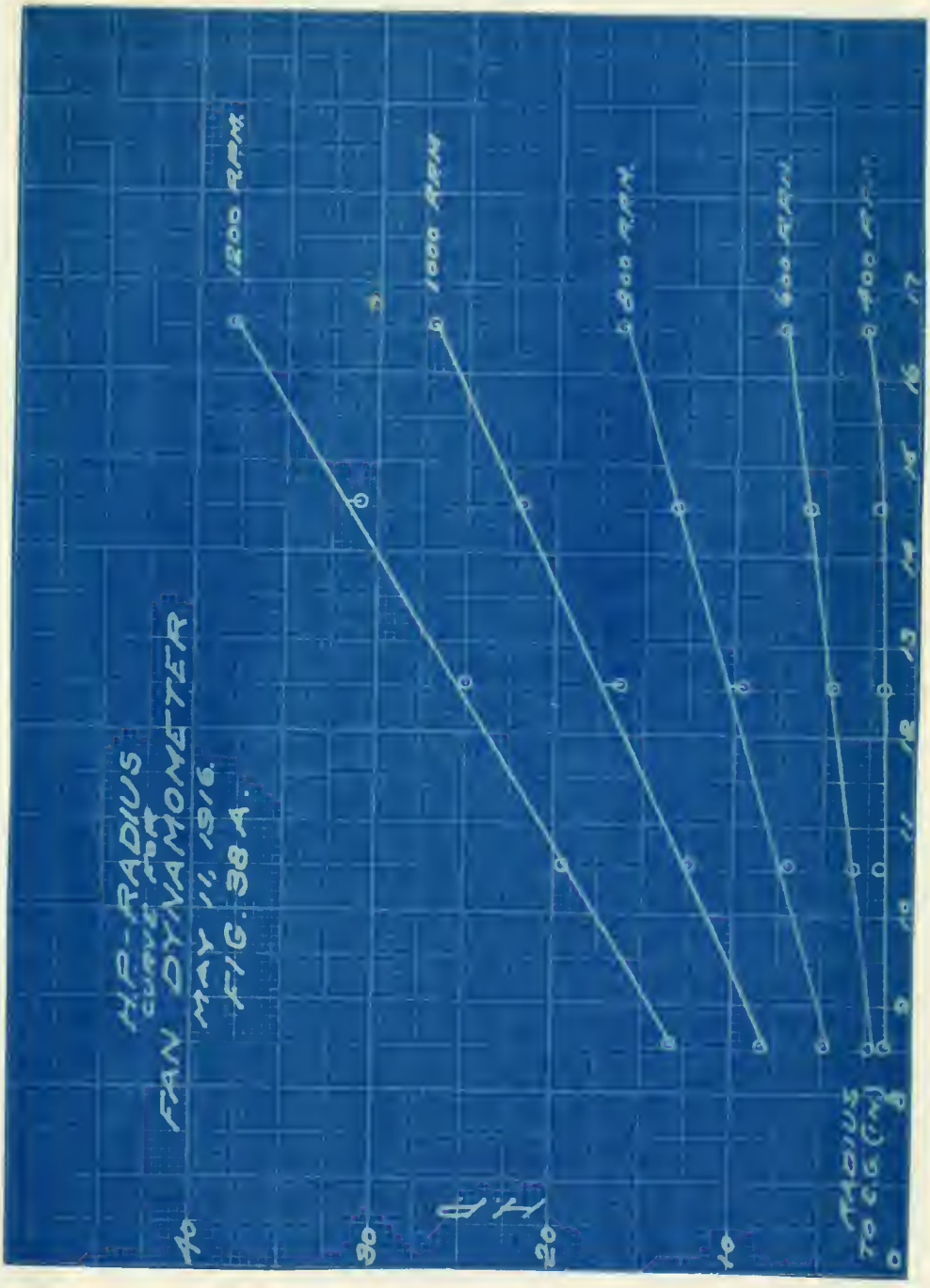
H.P.-RADIUS CURVES
FAN DYNAMOMETER
MAY 11, 1916.
FIG. 38 A.
1200 RPM.
1000 RPM.
800 RPM.
600 RPM.
400 RPM.
H.P.
RADIUS TO C.G. (IN.)
40
30
20
10
0

-:+:+:+:+:PART FOUR:+:+:+:+:+:-

C O N C L U S I O N.

C O N C L U S I O N

oCo

On account of a lack of time, this thesis was not made quite as extensive as planned. A delay in getting the cage made it impossible to run tests determining the effect of environment, and a lack of time was responsible that a greater number of blades were not made and tested out.

The object of the test was carried out in that a fan dynamometer was designed and built, and a test was made on it. This test showed that the horsepower varied with the pressure of the air against the fan blades, the barometric pressure, the temperature of the air, and the relative humidity. Two equations were evolved, one experimentally, which checked up with the tabulated results, and one theoretically and logically, which did not. It is probably impossible to evolve a theoretically sound equation which will check up

with actual results, on account of the great number of local factors which enter into the test, such as air currents and environment. In conclusion, it might be said that the fan dynamometer is very easy and simple to operate with the blades in any one position.

Its great operating disadvantage is the constant changing of the position of the blades. If an arm could be designed wherein the blades could be moved inwards or outwards while the dynamometer was in motion, the fan brake would be much improved. The fan dynamometer holds the engine down very well and causes it to run very sweetly and smoothly.

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